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CALLENDAR AND NICOLSON

ON THE

LAW OF CONDENSATION OF STEAM.

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XX

ON THE LAW OF
CONDENSATION OF STEAM

DEDUCED FROM MEASUREMENTS OF

TEMPERATURE-CYCLES OF THE WALLS AND STEAM IN THE CYLINDER OF A STEAM-ENGINE.

BY

HUGH LONGBOURNE CALLENDAR, M.A., F.R.S.,

AND

JOHN THOMAS NICOLSON, B.Sc.

WITH AN ABSTRACT OF THE DISCUSSION UPON THE PAPER.

EDITED BY

J. H. T. TUDSBERY, D.Sc., M. Inst. C.E.,
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THE INSTITUTION OF CIVIL ENGINEERS.

SECT. I.—MINUTES OF PROCEEDINGS.

30 November, 1897.

JOHN CLARKE HAWKSHAW, M.A., Member of Council, in the Chair.

(*Paper No. 3024.*)

"On the Law of Condensation of Steam deduced from Measurements of Temperature-Cycles of the Walls and Steam in the Cylinder of a Steam-Engine."

By HUGH LONGBOURNE CALLENDAR, M.A., F.R.S., and
JOHN THOMAS NICOLSON, B.Sc.

IN the discussion of steam-engine trials, it has been customary to compare the measured cylinder-feed with the quantity of steam indicated by the cards, with a view to deduce the so-called "missing quantity" of steam, and to arrive at the consequent loss of available energy resulting from condensation and re-evaporation. The question of cylinder condensation has also been elaborately discussed from a theoretical standpoint, on the assumption that the temperature-cycle of the metal surface is the same as that of the steam.

The object of the experiments recorded in this Paper, which were carried out in the summer of 1895, in the thermodynamic laboratory of the McDonald Engineering Building, at the McGill University, was the measurement of the cyclical interchange of heat between the walls and the steam. From a knowledge of the wall temperature-cycle, the amount of condensation taking place at a particular point under definite conditions can be readily ascertained, and the nature of the action can be studied separately, without confusion with other effects which are taking place in other parts of the cylinder at the same time. The mean temperature of the walls in different parts of the cylinder was also measured. A comparison of the results leads to a simple relation between the mean wall temperature, speed and condensation, which appears to hold through a considerable range of conditions.

The engine used in the experiments, Figs. 1, Plate 6, was made by the Robb Engineering Company, of Amherst, N.S. It has a large slide-valve with double ports, and a relief-back of the Porter type; and is designed to run at a speed of 250 revolutions per minute. In ordinary working, the speed is regulated by the governor on the fly-wheel, which automatically varies the travel and angular advance of the valve, in proportion to the load. For these experiments the governor was disconnected, and the valve-eccentric set to give the cut-off desired in each experiment. The principal dimensions are: Stroke, 12 inches; diameter of cylinder, 10·5 inches; piston displacement, 0·601 cubic foot; clearance volume, 0·060 cubic foot; slide-valve, 10·74 inches by 13·5 inches; steam-ports, 9·5 inches by 1·5 inch. In the present trials the engine was made single-acting by the addition of a brass lap 1·19 inch wide to the crank end.

CYLINDER-WALL TEMPERATURE-CYCLES.

The measurement of cylinder temperatures by means of mercury thermometers has been carried by Mr. Bryan Donkin[1] to a high degree of refinement. From a careful study of those experiments it was concluded that it would not be possible to obtain any more definitely quantitative results by the use of mercurial thermometers. The more laborious, but at the same time more powerful and pliable, methods of electrical thermometry were therefore adopted.

The Thermo-Electric Method.—This method has been applied by Professor E. Hall, of Harvard College;[2] but the Authors' results, obtained by a modification of that method, differ so considerably from his, that it is necessary to explain somewhat fully the points of difference between the two methods. In a thermo-electric circuit, the metal between the hot and cold junctions should be of continuously uniform quality, since different specimens even of the same metal are known to differ materially in thermo-electric power, and it was found necessary to take the greatest precautions to secure the most uniform quality in the materials used for the circuit. In the Hall method of inserting the thermo-junction, it would be a matter of some difficulty to secure absolute freedom from leakage. It is stated that leakage was noticed on several occasions, and it is evident that the intermittent rushes of steam through and about the junction might entirely change the temperature conditions. In order to avoid this source of trouble as far as possible, the junctions were made with the cast iron of the cylinder itself, holes being bored in the solid metal to within a carefully measured distance of the inner surface. This method has the further advantage of not introducing any foreign material into the thermo-electric circuit.

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[1] Minutes of Proceedings Inst. C E., vol. cvi. p. 264.

[2] Transactions of the American Institute of Electrical Engineers, 1891, vol. viii. p. 236.

In measuring the cyclical variations of temperature, the conditions of running should be the same when the observations are being taken at different points of the stroke, and at different depths in the metal. It is necessary to keep the engine running as steadily as possible, and to take the observations of the cycle in rapid succession. It is also important to take the indicator diagrams and the other observations of temperature simultaneously with the wall-cycle. A special cover was therefore cast for the cylinder, and fitted with eight thermo-couples at different depths. Any one or any two of these junctions could be connected to the galvanometer and the external cold junction, by a suitable arrangement of mercury-cups, and a contact-maker could be set to close the galvanometer circuit at any point of the stroke, and for any desired fraction of a revolution.

Since the observations had to be taken in rapid succession, the well-known compensation method of Poggendorf, in which the electromotive force of the thermo-couple is balanced against that of a steady current flowing in a uniform potentiometer wire, was adopted. The paraffin bath containing the external junction was maintained constantly at or very near 212° F., by a jacket of steam at atmospheric pressure. With a contact duration of only $\frac{1}{30}$ revolution, it was possible to read the galvanometer to nearly one-tenth of a degree.

Cycle Contact-Maker.—The cycle contact-maker, Figs. 2, Plate 6, closed the galvanometer circuit for a small fraction of a revolution at any desired point of the cycle. A pair of insulated revolving brushes were connected by a copper wire. One brush made contact with a central copper tube, the other with a number of copper sectors of different lengths let into the circumference of a wooden disk. A galvanometer circuit could thus be closed by any of these sectors, so as to alter the duration of the contact. The point of the cycle at which the brush made contact was determined by the setting of a divided circle. The contact-maker could be inserted in the circuit of either of two galvanometers, according as it was desired to observe a steam-cycle with the platinum thermometer, or a wall-cycle with one of the thermo-couples.

Experimental Cylinder Covers.—The cylinder-cover was provided with a steam-tight jacket, Figs. 3, Plate 6. Eight holes for the junction wires, arranged in a circle of 1·5 inch radius round the centre of the cover, were bored to depths of 0·01 inch, 0·02 inch, 0·04 inch, 0·08 inch, 0·16 inch, 0·32 inch, 0·64 inch, respectively, measured from the inner surface of the cover. The test wires were insulated with mica and india-rubber tube and washers, except at the bottom of the holes, where they made contact with the cast iron of the cover. In passing through the jacket, they were protected by steel tubes screwed and soldered into the metal of the cover. The central tube, seen projecting from the cylinder inside the jacket, was for clearance of the piston steam-thermometer to be subsequently described.

The Thermo-Couples.—Commercially pure nickel wire was first tried for the thermo-junctions, but the electromotive force of this impure nickel with cast iron was found to be three times less than with wrought-iron wire. This implied that pure iron wire would make with cast iron a much better thermo-junction than German silver, or even than pure nickel. It was, therefore, decided to use wrought-iron wire for the thermo-couples. Particular care was exercised in the choice of the materials. The iron wire used was from the same hank, and was carefully annealed after being bent into position. The cast-iron connections for the external junction were made by planing rods out of the middle of a 4-inch bar, cast at the same time and from the same ladle as the cylinder cover itself.

From experiments on the cylinder, the formula for E, the electromotive force of the cast-iron and wrought-iron junction, one of the junctions being at $t°$ C., and the other at 100° C., was found to be :—

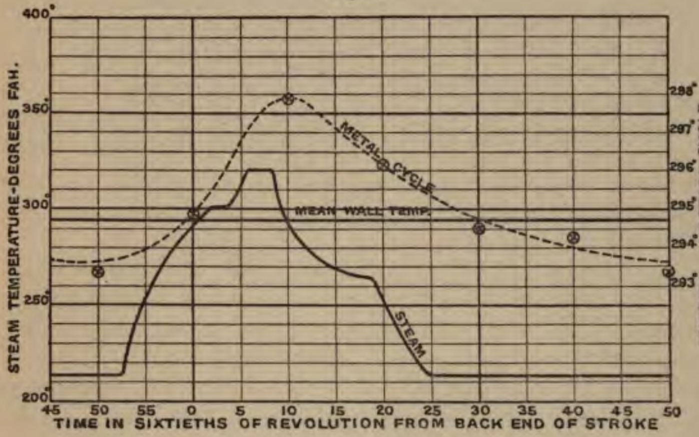
$$E = 1692 - 17 \cdot 86\,t + 0 \cdot 0094\,t^2, \text{ in microvolts.}$$

As a general rule, the thermo-couples were applied chiefly to the measurement of small differences of temperature at a known mean temperature $t°$ C. It was sufficient to use the simpler formula for the change of electromotive force per degree, namely :—

$$\frac{d\,E}{d\,t} = 17 \cdot 90 - 0 \cdot 0190\,t, \text{ microvolts per } 1° \text{ C.}$$

Observations of Wall-Cycles in the Cover.—The first observation of a wall-cycle is shown in *Fig. 5.* The full curve, corresponding to the scale on the left, gives the steam temperatures as deduced from the indicator-diagram taken simultaneously. The dotted curve, corresponding to the scale on the right, gives the variations of temperature observed at a depth of 0·01 inch in the cover, at a speed of 100 revolutions per minute. The observations are shown at (×). The cycle shows a range of only 4·3° F., corresponding to an absorption of 1 thermal unit F. per square foot per cycle, or a condensation of about $\frac{1}{1000}$ lb. of steam. The same observation was repeated, when a range of about 4° F. was again found.

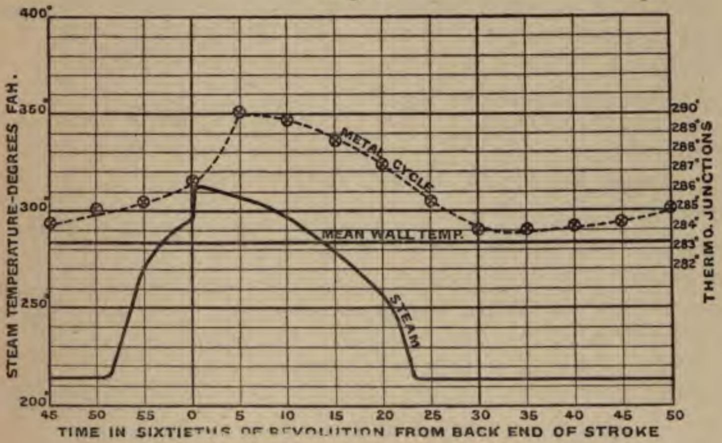
Fig. 5.



Speed, 100 revolutions per minute.

Metal Cycle at $\frac{1}{100}$ Inch Cover.

Fig. 6.



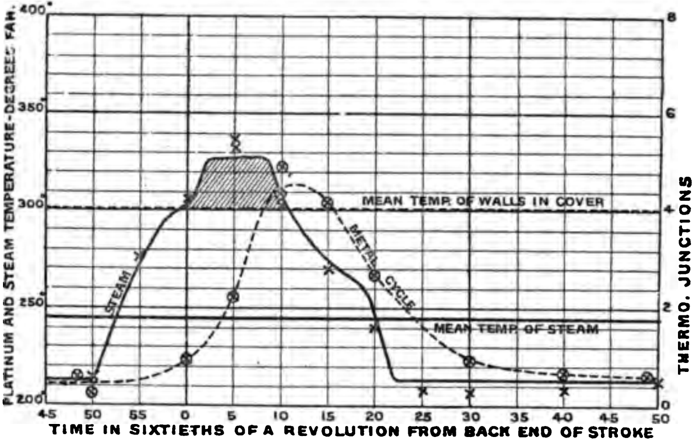
Speed, 46 revolutions per minute; throttled.

Metal Cycle at $\frac{1}{25}$ Inch Cover.

In taking the observations for the wall-cycle, shown in *Fig. 6*, the differential method was adopted for the first time, which accounts for the greater smoothness of the curve.

In this method the thermo-electric circuit consisted simply of a small portion of the cast iron of the cylinder and of two iron wires making contact at different depths in the metal, at points which are at nearly the same mean temperature. Only a small difference of temperature, for which the thermo-electric method is admirably suited, is therefore observed. By this method, nearly all the troubles that otherwise arise from the slow changes of conditions always in progress are avoided, since a gradual change of temperature will affect the metal at a depth of ½ inch to the same extent as the surface of the metal; whereas the rapid cyclical changes are practically evanescent at the greater depth. For the remainder of the experiments the differential method was always used for the wall-cycles. The cycle shown in *Fig. 7* is an illustration of an observation taken about a month later, at a depth of 0·039 inch, and a speed of 73·4 revolutions per minute.

Fig. 7.



Speed, 73·4 revolutions per minute.

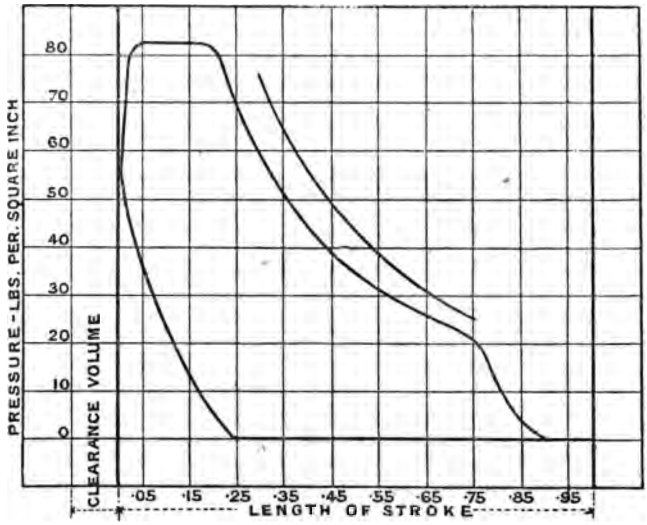
Metal Cycle at $\frac{1}{25}$ Inch Cover.

The shaded area of the steam curve, above the line representing the mean temperature of the cover, may be called the "Condensation Area," as it appears to determine the amount of condensation taking place. For the cycles shown in *Figs. 5* and *6*, the test-wires were pressed against the metal of the cover. For

Fig. 7 and subsequent observations, the apparatus had been entirely rearranged. The holes had been bored and measured afresh, new wires of different lengths had been fitted, and all the joints and thermo-junctions had been carefully soldered with pure tin. This insured a good and permanent contact, but otherwise no difference in the observed temperature cycles could be detected. *Fig. 8* shows an average card belonging to this trial, from which the cycle curve of *Fig. 7* was deduced. Several trials were run with this valve-setting.

Wall-Cycles on Barrel-Surface at Side.—A set of eighteen thermo-junctions were fitted along the side of the cylinder to observe the wall-cycles and the distribution of temperature along the barrel-surface. The arrangement of the side junctions is shown in Figs. 4, Plate 6. To facilitate making and changing connections, the majority of the test-wires were connected to mercury-cups in a slip of wood fixed to the cylinder inside the cast-iron lagging. At the back end of the stroke, and at 4 inches, 6 inches and 12 inches along the side from it, pairs of junctions were inserted, one bored to leave 0·04 inch of metal, and the other ½ inch at each point. The remainder of the junctions, at 2 inches, 8 inches, 10 inches, 14 inches and 16 inches respectively, were

Fig. 8.



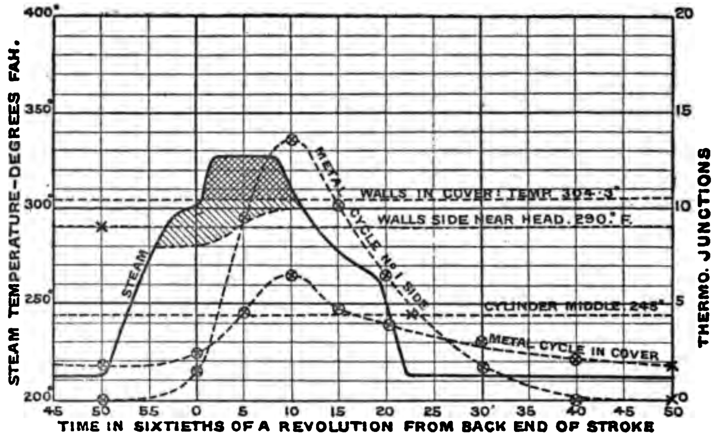
Speed, 73·4 revolutions per minute; cut-off 0·200.

INDICATOR DIAGRAM.

holes bored to the depth of ½ inch, into which iron wires could be inserted for observing the length-distribution of temperature. There were four similar holes at points above and below on a line round the middle of the cylinder. Three vertical holes, 2 inches deep, were also bored in the metal of the side, at distances of 1 inch, 7·5 inches and 15 inches along the stroke, for the insertion of mercury or platinum thermometers, for calibrating the side junctions, and for observing the mean wall temperatures.

Comparison of Cycles on Cover and Side.—The curves in *Fig. 9* illustrate cycles observed in the same trial. The mean temperature of the side walls opposite the clearance space was 14·3° F. lower than that of the cover. Junction No. 1, side, was situated at this point at the back end of the stroke, and was exposed to the same steam as the cover. The ranges observed, at the cover and side, were 4·9° F. and 13·5° F. respectively; the speed being 44 revolutions per minute. The depths of the junctions were, cover, 0·039 inch, and side, 0·037 inch respectively. This particular cycle at the side was the largest observed throughout the trials. It corresponds to a range of about 20° F. at the surface of the metal. The "condensation area" for each junction is shaded as in the previous example. It will be observed that the temperature of the wall begins to rise near the end of compression, shortly after condensation begins, and reaches a maximum shortly after cut-off near the end of the condensation period. The fall due to

Fig. 9.



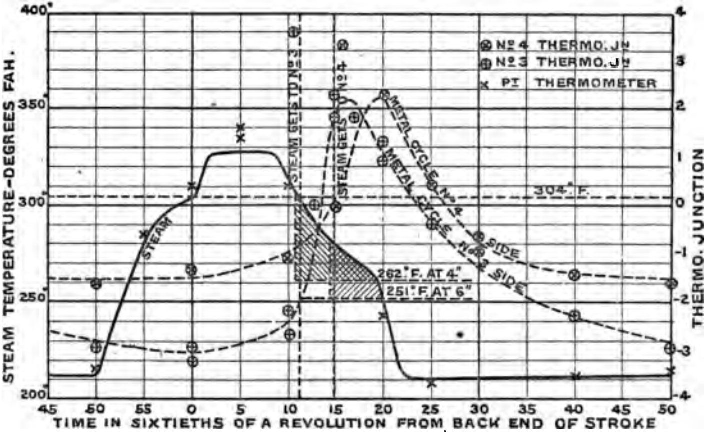
Speed, 43·9 revolutions per minute.

METAL CYCLES $\frac{1}{25}$ INCH IN COVER AND No. 1 SIDE.

re-evaporation is nearly as rapid as the rise. In drawing the lower boundary of the condensation area for this cycle, the probable surface temperature is taken instead of the mean; but the difference of area is small. The magnitude of the condensation area would be little affected by taking the mean temperature instead of the surface temperature, but the cycle curve would not then correspond so well, allowing for lag, with the probable rate and epoch of condensation.

Side-Wall Cycles beyond Cut off.—*Fig. 10* gives an example, on a temperature scale two and a half times as large, of cycles observed on junctions Nos. 3 and 4, at 4 inches and 6 inches along the cylinder from the back end. The lower observations at the points

Fig. 10.



Speed, 45·6 revolutions per minute.

METAL CYCLES IN NOS. 3 AND 4 SIDE.

0, 10, 15 and 20, for No. 3, were taken after the observation of cycle No. 4, and illustrate the order of accuracy attainable in the measurement of these cycles. It will be noticed that the temperature begins to rise at each point before steam reaches it, if the piston is assumed to fit accurately. This may be explained partly by the friction of the hot piston, partly by a probable small leakage of steam, and partly by the fact that the piston-rings are about ½ inch from the face of the piston. In estimating the condensation areas for these junctions, ½ inch is allowed for the imperfect fit of the piston; that is to say, it is assumed that the steam reaches the junction when the face of the piston is ½ inch behind the corresponding point of the stroke. The thickness of metal at these junctions was 0·037 inch and 0·039 inch respectively, and is probably accurate to $\frac{1}{500}$. The speed was 45·6 revolutions per minute.

Effect of Later Cut-off.—A few trials only were run at a later cut-off—three at one-third, and two at one-half. The data of these trials were not in general sufficiently complete to afford a satisfactory basis of comparison. With a later cut-off the range of the wall-cycles observed on the cover did not differ materially from those previously recorded. The longer steam contact was compensated by a higher wall temperature. On the side wall, near the back end of the cylinder, the rise of temperature was nearly 20° F. at one-half cut-off as compared with one-fifth. The range of the cycle was reduced from 11·0° F. to 9·2° F., at a depth of 0·037 inch in the metal, and a speed of 49 revolutions per minute. At 4 inches and 6 inches along the cylinder the ranges of the wall-cycles, at the same depth and cut-off, were increased to 7·2° F. and 5·0° F. respectively, as the full-pressure steam reached these points of the wall. The temperature of the middle of the cylinder was raised nearly 30° F. as compared with one-fifth cut-off, but the comparison could not be made quite satisfactorily owing to slightly different conditions of steam pressure. The total condensation, including the later portions of the stroke, was probably increased somewhat, but at the same time the cylinder feed was more than doubled.

TEMPERATURE DISTRIBUTION AND STEADY HEAT FLOW.

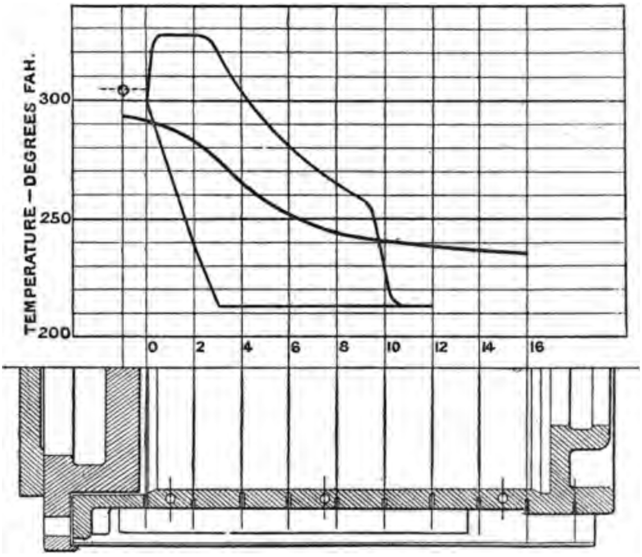
Outward Temperature Gradients.—Careful measurements of the temperature gradients were made in various parts of the cylinder with a view to deduce the steady flow of heat. From the mean of several observations in the thickness of the cover, which was protected by an air-jacket, a probable gradient of 0·55° F. per inch was deduced, a value which corresponds fairly well with the probable external loss. At the points 4 inches and 6 inches along the side, the temperature of the inner surface was 2·4° and 1·1° F. respectively lower than that of the outer surface. This curious and at first sight paradoxical result, means that the cylinder wall at these points was losing heat to the steam.

Temperature Gradients on Barrel Surface.—The longitudinal distribution of temperature for one-fifth cut-off, single-acting, is shown in *Fig. 11*, together with a corresponding diagram of steam temperatures deduced from the average indicator-diagram, with a mean speed of 45·6 revolutions per minute.

On several occasions special experiments showed that a change of speed produced little change in the distribution of temperature. The mean temperature at the centre of the cover in the same trials is shown by the mark (–), at 305° F., opposite the section of the cover.

Effect of Longitudinal Conduction, and Piston Convection.—The greater part of the condensation on the barrel-admission surface in a small single-acting engine is probably due to the lowering of temperature caused by conduction and convection along the cylinder. In this set of observations the maximum gradient of 9·3° F. per inch occurs at a point a little after cut-off, and corresponds to a loss of heat by the admission surface of 11·2 thermal units per minute. The cyclical convection of heat by the piston is a factor which may be of some importance in a single-acting engine. It will depend on the surface of the piston, on the closeness of the contact, and on the difference of temperature between the ends of the cylinder. In the present case the mean difference of temperature was 45° F. at one-fifth cut-off, and nearly 65° F. at one-half cut-off. The curved surface of the piston was approximately 1 square foot. The effect of piston convection may be traced in the form of the cycles observed at one of the side junctions. In a double-acting engine, where both ends of the

Fig. 11.



SECTION OF CYLINDER AND COVER, SHOWING POSITION OF THERMO-JUNCTIONS AND THERMOMETERS.

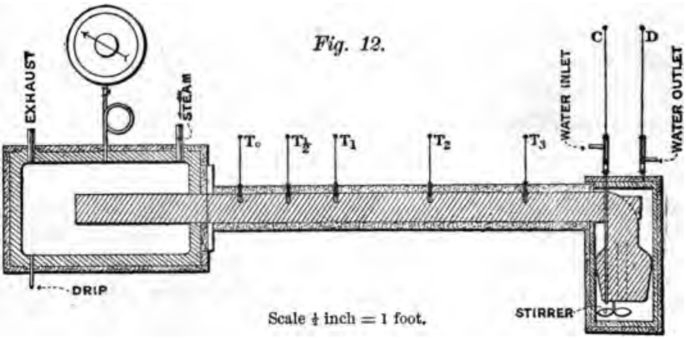
cylinder are at nearly the same temperature, the effect of piston convection would be of very little importance. In a high-speed single-acting engine like the Willans engine, it is probable that piston convection might be the most important factor in cooling the barrel-admission surface, and causing initial condensation.

Abstraction of Heat by the Condensation of Wet Steam.—For a distance of nearly 6 inches along the side, while the engine was running at one-fifth cut-off, the external surface was nearly 1° F. on the average hotter than the internal. Heat, supplied by conduction, was being abstracted from the inner surface at a rate of at least 5 thermal units per square foot per minute. On stopping the engine the temperature along this belt immediately began to rise, and continued rising for some minutes. The heat abstracted by evaporation is greater than that supplied by condensation over this part of the surface. The probable explanation is to be found in the wetness of the steam due to adiabatic expansion. (See p. 41.)

CONDUCTIVITY AND SPECIFIC HEAT OF CAST IRON.

The electrical resistance of cast iron was found to be nearly ten times greater than that of wrought iron. Considerable difference in the thermal conductivity was therefore to be expected. Observations on the cylinder showed that the conductivity of cast iron was probably some 30 per cent. less than the value generally assumed. It was therefore desirable to attempt a special determination of this important constant by the most accurate methods. With this object the apparatus shown in *Fig. 12* was designed. The metal used was a 4-inch bar of iron, cast from the same ladle as the cylinder cover.

Fig. 12.



APPARATUS FOR DETERMINING THE THERMAL CONDUCTIVITY OF CAST IRON.

Two independent methods were employed—(1) The calorimetric method, in which the quantity of heat transmitted is directly measured; and (2) that of Ångström, which depends on observation of the propagation of temperature waves.

The Calorimetric Method.—In terms of the units employed, the result of this method may be stated as follows:—The quantity of heat conducted across a plate of cast iron 1 foot square and 1 inch thick, for a difference of temperature of 1° F. between its faces, amounts to 5·65 thermal units (pound-degree-F.) per minute, the plate being at a temperature of 104° F. (The value generally assumed for wrought iron is 7·5 in the same units.) Expressed in terms of C.G.S. units centigrade, the values become, cast iron, 0·117, wrought iron, 0·155.

The Ångström Method.—The diffusivity or thermometric conductivity is the ratio of the calorimetric conductivity k to the thermal capacity c of unit volume. The unit volume was taken to be a plate 1 foot square and 1 inch thick, in order to harmonize with the units employed for the calorimetric conductivity. Observations were taken at mean temperatures of 215° F. and 130° F., from which was derived for the probable variation of the diffusivity with temperature the formula:

$$\frac{k}{c} = 1 \cdot 42 - 0 \cdot 0010\, t,$$

where t is the temperature in degrees Fahrenheit.

Specific Heat of Cast-Iron.—From a specially devised series of experiments there was deduced for the specific heat s of the specimen of cast iron at a temperature t° F., the formula:

$$s = 0 \cdot 1090 + 0 \cdot 000060\, t,$$

which is probably correct to 1 per cent. between 200° and 350° F. Combining this with the result previously given for the diffusivity, the following values for the thermal capacity c of a plate 1 foot square and 1 inch thick, and for the calorimetric conductivity k are obtained:—

TABLE I.—CONDUCTIVITY AND SPECIFIC HEAT OF CAST IRON.

Temperature.	Specific Heat s.	Thermal Capacity c.	Diffusivity $\frac{k}{c}$.	Conductivity k.
° F.				
100	0·115	4·21	1·32	5·55
150	0·118	4·32	1·27	5·48
200	0·121	4·43	1·22	5·40
250	0·124	4·54	1·17	5·31
300	0·127	4·65	1·12	5·21
350	0·130	4·76	1·07	5·10

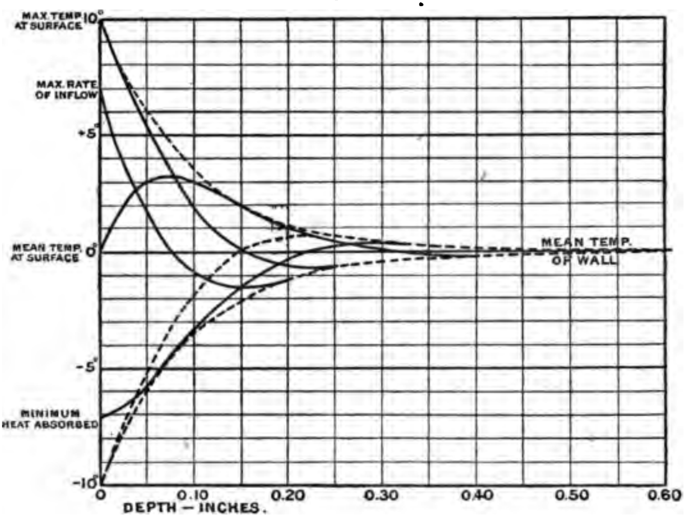
Density, Thermal Capacity and Composition of the Bar.—The density of the bar gives a weight of 36·6 lbs. for a plate 1 foot square and 1 inch thick. The values for c given in Table I, are found by multiplying the specific heat s by 36·6. It will be observed that the values of the thermal constants of cast iron vary to a considerable extent with the temperature. For calculating the Tables it is necessary to assume certain average values of these constants and the numbers, $k = 5 \cdot 4$, $c = 4 \cdot 5$, $\frac{k}{c} = 1 \cdot 20$ are taken, which correspond to a temperature of 220° F. The composition of the bar is shown by the appended chemical analysis.

	Per cent.
Total carbon	3·08
Graphite . .	2·86
Silicon . .	2·89
Phosphorus .	1·05
Manganese .	0·85
Sulphur . .	0·022
Total .	7·89

CONDENSATION.

Cyclical Heat Absorption.—The curves given in *Fig. 13* illustrate the cyclical absorption of heat by the metal of a cast-iron cylinder at a speed of 42 revolutions per minute, for a simple harmonic variation of surface temperature with a range of 20°. The full curves show the simultaneous values of the temperature at

Fig. 13.



Speed, 42 revolutions per minute; range, 20° at surface.

TEMPERATURE DEPTH CURVES; CAST-IRON CYLINDER.

different depths for four typical points of the cycle. The heat absorbed or rejected by the metal between any two points of the cycle is proportional to the area included between the corresponding temperature-depth curves. The dotted boundary curves have the equation $t = \pm e^{-mx}$, and show the rate of diminution of the range of temperature t with increase of depth x. The index coefficient m is given by the formula $m = \sqrt{\frac{\pi n c}{k}}$, where n is the number of revolutions per minute. In the case figured, $m = 10\cdot 5$. The wave-length of the temperature-oscillation is $\frac{6}{10}$ inch. At this depth the retardation amounts to one complete period, and the range is reduced to less than one five-hundredth part of its value at the surface. The wave-length in each case may thus conveniently be regarded as the practical limit of penetration of the heat-waves.

Numerical Values for Cast Iron.—The following Table has been calculated for a cast-iron cylinder, with a simple-harmonic cycle of 10° F. range at the surface, at various speeds, assuming the values $k = 5\cdot 4$, $c = 4\cdot 5$, $\frac{k}{c} = 1\cdot 20$, in the units given. The values for any other range are directly proportional to the range. For calculating the various columns the numerical formulas used are:—

Index coefficient $m = 1\cdot 618\ \sqrt{n}$. Wave-length $= \frac{6\cdot 28}{m}$.

Temperature range at 0·040 inch depth $= 10° \times e^{-0\cdot 040\,m}$.

Heat absorbed in thermal units Fahr. per square foot per cycle $= 10° \times \frac{3\cdot 18}{m}$.

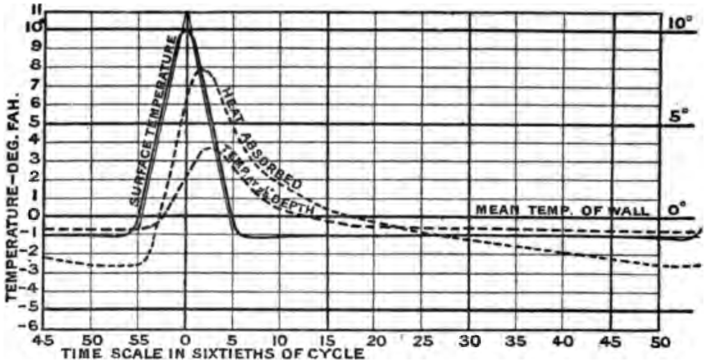
TABLE II.—CYCLICAL HEAT ABSORPTION AT DIFFERENT SPEEDS IN CAST IRON.

Revolutions per Minute.	Index Coefficient.	Wave-Length (Penetration).	Temperature Range at 0·040 inch.	Heat Absorbed in Thermal Units per Square Foot.	
n.	*m.*	Inch.	Per 10° Surface.	Per Revolution.	Per Minute.
25	8·09	0·777	7·25	3·93	98
40	10·24	0·613	6·63	3·11	124
50	11·44	0·549	6·32	2·78	139
60	12·54	0·501	6·05	2·53	152
70	13·54	0·464	5·82	2·35	164
80	14·47	0·434	5·60	2·20	176
90	15·35	0·409	5·41	2·07	187
100	16·18	0·388	5·24	1·97	197
150	19·82	0·317	4·52	1·61	242
200	22·88	0·275	4·00	1·40	280
300	28·02	0·224	3·26	1·14	342
400	32·36	0·194	2·75	0·98	392
500	36·18	0·174	2·35	0·88	440

The above Table is used in calculating the results of the wall-cycles.

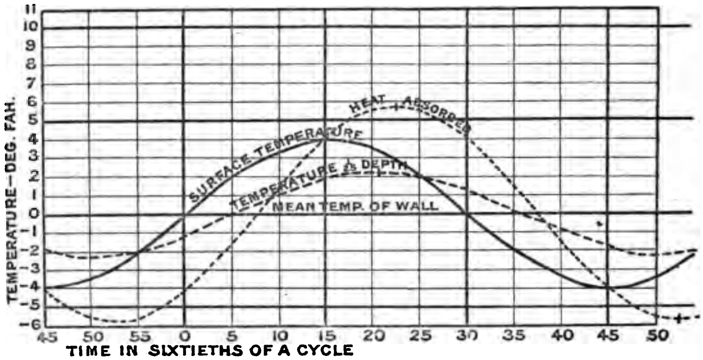
Effect of the Form of the Cycle.—The formulas given, refer to a simple-harmonic wave, which is propagated without change of form. Cycles of any other form must be analyzed by the Fourier method into their simple-harmonic components. In order to compare the results of assuming an entirely arbitrary and peculiar cycle, instead of the simple-harmonic, the coefficients of the first twelve terms of the series, representing the triangular cycle shown in *Fig. 14*, were calculated by the Fourier method. This series, when plotted, gives a curve, *Fig. 14*, rising to a maximum

Fig. 14.



TRIANGULAR CYCLE FOR A RANGE OF 4·5° AT $\frac{1}{25}$ INCH IN METAL.

Fig. 15.



SIMPLE-HARMONIC CYCLE FOR A RANGE OF 4·5° AT $\frac{1}{25}$ INCH.

of 10° at the point O, and giving a very nearly constant temperature of $-1°$ for $\frac{5}{6}$ths of the cycle. The corresponding curve of heat-absorption, and the curve of temperature at a depth of 0·040 inch, may be compared with the simple-harmonic cycle, *Fig. 15.* The temperature range of *Fig. 14* at a depth of 0·040 inch is found to be 4·5°, corresponding to a surface range of 11°, at a speed of seventy-seven revolutions in cast-iron. For the same range at a depth of 0·040 inch, the simple-harmonic surface-range is 8° only. The surface-ranges differ considerably, but the quantities of heat absorbed are nearly equal.

Table III.—Results of Observed Wall-Cycles.

Trial Mark and Cut-off.	Revolutions per Minute.	Thermo-Junction Depth.		Temperature Ranges.		Heat (Q) Absorbed per Square Foot per Cycle.	Maximum Card Temperature.	Mean Wall Temperature.	Condensation Area (A), Degree-Seconds per Cycle.	Ratio, $\frac{Q}{A}$.
				Observed at Junction.	Calculated for Surface.					
		Ins.	Ins.	°	°		°	°		
I$\frac{1}{5}$	100·0	0·010 C		4·3	5·1	1·00	318	295	1·50	0·67
II$\frac{1}{5}$	102·0	0·010 C		3·8	4·5	0·88	319	296	1·45	0·61
VIII$\frac{1}{3}$	46·0	0·040 C		6·1	9·4	2·75	313	287	4·28	0·64
X$\frac{1}{3}$	77·0	0·037	S 0	11·0	18·9	4·25	317	277	5·60	0·76
XVI$\frac{1}{5}$	73·4	0·039 C		4·0	6·8	1·56	324	301	2·58	0·61
XVII*a*	70·4	0·039 C		4·0	6·7	1·57	335	310	2·64	0·60
XVII$\frac{1}{5}$	70·4	0·013 C		5·6	6·6	1·55	335	310	2·61	0·59
XVII*b*	97·0	0·039 C		3·3	6·2	1·26	329	305	1·90	0·66
XVIII	45·6	0·039	S 6	3·5	5·4	1·56	329	251	2·43	0·64
XVIII	45·6	0·037	S 4	4·7	7·2	2·07	327	262	3·45	0·60
XIX	43·8	0·039 C		4·9	7·5	2·20	329	305	4·20	0·53
XIX	43·8	0·037	S 0	13·5	20·2	6·00	328	291	9·13	0·65
XX*a*	47·7	0·039 C		4·6	7·2	2·05	331	307	3·80	0·54
XX*a*	47·7	0·037	S 0	11·0	17·2	4·90	331	293	7·65	0·64
XX*a*	47·7	0·037	S 4	4·3	6·7	1·91	331	265	3·31	0·58
XX*b*	81·7	0·039 C		3·4	6·0	1·31	331	306	2·32	0·56
XX*b*	81·7	0·037	S 0	8·3	14·2	3·10	330	292	4·60	0·67
XX*b*	81·7	0·037	S 4	3·5	6·0	1·31	331	264	2·03	0·65

The above Table includes all the observations of wall-cycles, for which complete data were available. In the third column C stands for cover, and S for side. For the latter junctions, the distance in inches along the side is also given. In trial X, the cycle at S, 0 inch, was taken against one of the junctions in the cover. The maximum card temperature is deduced from an indicator-diagram taken in the middle of the cycle. The mean wall temperature is that at the middle of the wall at the position of the junction, from observations taken before and after the cycle. These temperatures are probably right to 0·5° F.

Relation between Temperature, Speed and Condensation.—A comparison of the temperature-ranges at the surface of the metal, with the steam-cycles and the mean wall temperatures, appears to demonstrate that, even at the lowest speed of these trials, and making allowance for the form of the surface cycle, the time does not suffice for the steam to raise the temperature of the surface of the wall more than a small part of the way up to its own temperature. The largest surface-range of 20° F., observed in Trial XIX at forty-four revolutions, with a condensation lasting for nearly $\frac{1}{3}$ second, would raise the surface to 301° F. only, the temperature of the steam being 328° F. during the latter half of the interval. It is hardly possible to suppose this resistance to the passage of the heat from the steam to the metal, to be due to the existence of a surface film of oil or water. Assuming such a film to have a conductivity only one-hundredth of that of iron, it would require a thickness of at least 0·020 inch to produce the observed effects, if the surface of the film itself were instantly raised to the steam temperature. The cylinder was frequently examined immediately after a run, but the interior was invariably found clean and nearly free from grease. The grease film was certainly less than one-thousandth of an inch in thickness. With regard to water, the case is perhaps stronger. The maximum observed absorption of 6 T.U. per square foot, would correspond to the condensation of a film about one-thousandth of an inch thick. If the resistance is due to a water film, the evaporation must be incomplete, and a film must remain from stroke to stroke. Further, this must also take place so uniformly and consistently as to give perfectly regular resistance in all parts of the cylinder, and throughout the trials. The steadiness and consistency of the readings are perhaps the best proof, but there is other strong evidence, that such a film was not present.

The obvious inference from the observations is that the rate of condensation of steam is physically limited, and it is necessary to assume a provisional law of condensation in order to compare the results. As a simple and workable hypothesis, and for other reasons, the rate of condensation of steam on a metal surface was assumed to be proportional to the difference of temperature, and to be independent of the pressure. This assumption would make the amount of condensation taking place on any part of the walls, proportional per cycle to the average excess of temperature of the steam, multiplied by the time during which the temperature is above that of the walls. This product is found by measuring the condensation area on the cycle diagram included between the curves representing the steam and the wall-surface cycles. It is conveniently measured by the product of the time in seconds into the average difference of temperature in degrees. If this area on the cycle-diagram is measured in degrees of temperature, and in sixtieths of a revolution, the result gives the condensation area in degree-seconds per minute. This result divided by the revolutions per minute gives, further, the condensation area in degree-seconds per cycle. These condensation areas have been measured for each of the observed wall-cycles, and are given in the last column but one of Table III. The last column gives the ratio of the heat Q absorbed per cycle to the condensation area A in each case. The approximate constancy of this ratio would appear to indicate that the hypothesis is at least a first approximation to the truth.

Method of Estimating the Total Condensation at any Epoch.—It is evident from Table III that the condensation area is fairly proportional to the amount of condensation taking place at any point over the range covered by the experiments. It is further assumed, if a vertical line is drawn on the diagram of the cycle corresponding to the position of the piston at any epoch, the condensation areas measured to that line may be taken as proportional to the condensation which has taken place at each point of the surface up to the epoch considered. It is possible in this manner to arrive at a fair estimate of the amount of condensation at or shortly after cut-off, before re-evaporation has commenced. There is, further, evidence that the re-evaporation follows the same law, and may be in many cases treated in a similar manner.

Numerical Application of Method of Condensation Areas.—To make an estimate of the condensation in Trials XVI-XX, at 0·250 of the stroke, shortly after cut-off, and at 0·700 of the stroke, shortly before release, the mean wall temperatures are assumed to be independent of the speed, provided that there is no wire-drawing or other change of conditions. Hence the condensation areas, measured on the cycle-diagram in degrees of temperature and in sixtieths of a revolution, are the same or nearly so at different speeds. It is sufficient, therefore, for this estimate to take the area so measured and multiply it by 0·61, the mean of the values of the ratio $\frac{Q}{A}$ in Table III, in order to deduce the cyclical heat-absorption for the parts of the surface considered.

The following Table gives the total heat-absorption on the clearance surfaces. The portions of the barrel surface subsequently exposed by the motion of the piston, require to be treated somewhat differently. The temperature of the piston was specially determined by means of a platinum thermometer inserted through a hole in the piston-rod.

TABLE IV.—CYCLICAL HEAT-ABSORPTION FOR CLEARANCE SURFACES.

Portions of the Surface Considered	Area of Surface.	Mean Temperature.	Condensation Area per Square Foot per Minute.	Heat Absorbed in Thermal Unit Fahr. per Minute.
	Square Feet.	° F.	° Seconds.	
Cover, face 10·5 inches diameter .	0·60	305	185	68
„ side 3·0 „	0·70	305	185	79
Piston, face 10·5 inches diameter .	0·60	295	300	110
„ side 0·5 „	0·11	295	300	20
Barrel, side 3·0 „	0·71	297	285	123
Counterbore 0·5 „	0·12	291	380	28
Ports and valve . „	0·90	305	185	102
Sums and Means . .	3·74	301	231	530

At 0·250 of the stroke, 3 inches of the barrel surface have been exposed by the motion of the piston. Estimating the total contribution of this portion of the surface at 55 T.U.F. per minute up to 0·250 of the stroke, there is a total heat-absorption of 585 T.U.F. per minute at 0·250 of the stroke, which corresponds approximately under the conditions of the trials, to the condensation of 0·65 of a pound of steam per minute. The condensation per cycle at this point for any of the trials considered, may be obtained by dividing this result by the revolutions per minute. It will be observed that the clearance surfaces contribute about 90 per cent. of the total condensation.

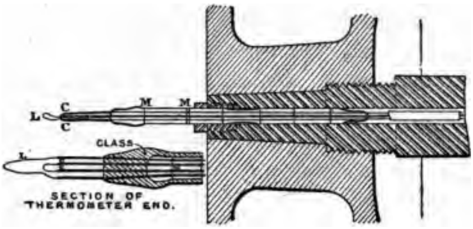
Estimate of Re-evaporation during Expansion.—It is not possible to make an estimate of re-evaporation with the same degree of probability as condensation, but some idea may be gained, by a similar method, of the amount of re-evaporation that has taken place before release. A probable excess of re-evaporation over condensation equivalent to nearly 300 T.U.F. per minute is found, and the quantity of heat rejected by the metal at release and during the exhaust period, generally called the exhaust waste, may be estimated at 250 T.U.F. per minute, under the conditions of the present trials.

STEAM CYCLES.

The Steam Thermometer.—The thermometers, which were sufficiently sensitive to follow the changes of temperature of the steam throughout the stroke, were made of very fine platinum wire, $\frac{1}{1,000}$ inch in diameter. The method of attachment to the piston is shown in *Fig. 16.*

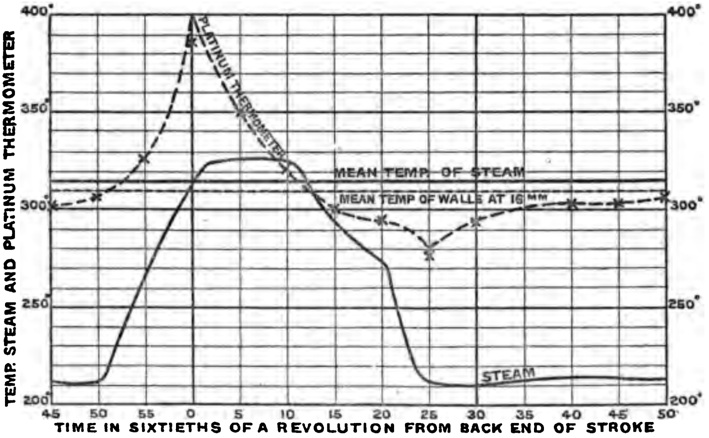
Temperature Cycle of Steam in Cover.—The first set of observations with one of these thermometers was made with the instrument fixed in a hole in the cylinder-cover. The results are exhibited in *Fig. 17.* The indicated steam temperatures are shown by the full curve, the temperatures of the platinum thermometer by the dotted curve. The temperature scale is the same for both. The most striking feature of the platinum curve is the great superheating shown at the end of compression. During admission the temperature rapidly falls. At, and shortly after, cut-off the thermometer invariably showed a temperature 2° or 3° below the

Fig. 16.



Section of Piston showing Platinum Thermometer.

Fig. 17.



Platinum Thermometer in Steam in ⅜-inch Hole in Cover.

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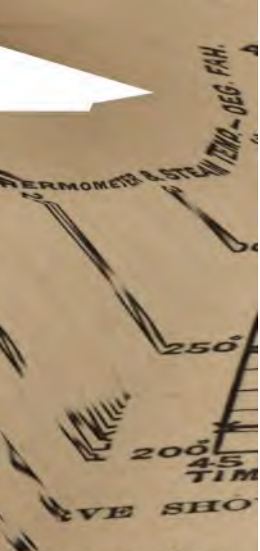

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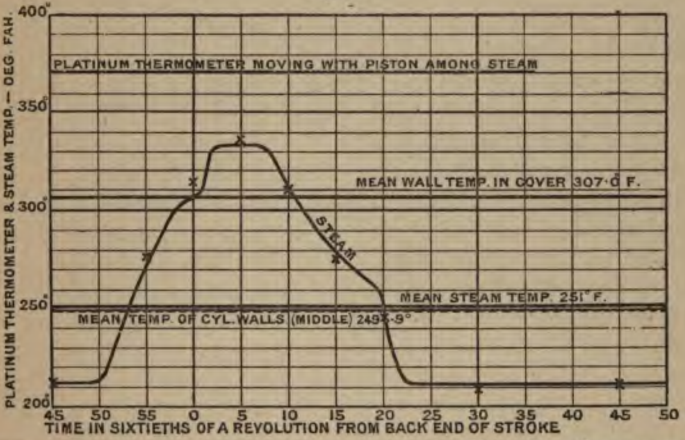
indicator. It soon, however, rises above the indicated curve, and, with the exception of a sudden drop at release, due to adiabatic expansion, continually approaches the temperature of the walls during the exhaust period. The actual superheating of the steam during compression must have been greater than that shown by the thermometer for two reasons. The method of observation gives the mean temperature during a certain interval of contact, generally $\frac{1}{30}$ revolution, and cannot therefore reproduce a very short and sharp maximum. Secondly, although the thermometer was certainly very sensitive, the lag must have been appreciable on a rise of 100° F. taking place in 0·1 second. It is remarkable that the effect of radiation from the cool surrounding walls is not more noticeable. To test the effect of pure radiation, as compared with that of convection, on these thermometers, a special experiment was made, when it was found that the rate of loss of heat by pure radiation for this very fine wire, at temperatures between 200° F. and 350° F., was between fifty and one hundred times less than that due to convection. The possible error due to direct radiation from the surrounding walls does not, therefore, amount to more than 1° or 2°, and the thermometer is really indicating the temperature of the steam around it. The peculiar characteristics of the platinum curve in *Fig. 17* were verified on several occasions, and with different settings of the valve, the results observed in every case being similar.

Piston Steam Thermometer.—To observe the temperature of the main body of the steam at a distance from the walls, a similar thermometer was attached to the piston in the manner shown in *Fig. 16.* The thermometer projected from the piston for a distance of about 3 inches, and was received at the back end of the stroke in a tube 1 inch in diameter in the centre of the cover. The indications of this thermometer at different speeds and at different settings of the valve, were in remarkably close agreement with the card. Systematic differences, however, were always observed, which, from their consistency and from the great number of observations, cannot be attributed either to errors of the indicator or of the thermometer. The curve in *Fig. 18* is drawn from the card taken simultaneously with the observations of the platinum thermometer. The differences observed are almost too small to be shown. They are much greater, however, than the uncertainty of the observations themselves. The temperatures at admission and release were much steadier than in the cover. At most other points of the stroke it was possible to take readings to 0·1 degree, and the extreme variations often did not amount to more than 0·5 degree for several minutes. The superheating shown during compression was small; the greatest amount was shown in the middle of the admission period. The centre of the steam appears to have been at this epoch generally 4° F. or 5° F. above the indicator. *Fig. 18* shows the smallest value recorded, namely, 2° F. Other examples will be found in *Fig. 7* and *Fig. 10.* Throughout the expansion curve, the readings of the platinum thermometer were between 2° F. and 3° F. lower than the indicator. At the end of the stroke, and for part of the exhaust period, the temperature fell to between 207° F. and 208° F., but recovered to 212° F., or close to the barometric temperature, before the end of exhaust. This may have been due to a real lowering of pressure, owing to the rapidity of condensation in the surface condenser at a pressure slightly below that of the atmosphere. The corresponding difference of pressure is only 1 lb. The lowering of temperature of the steam during expansion appears, on consideration, to be too large and regular to be explained by any error or lag of the indicator; lag of the thermometer would have the opposite effect.

Fig. 18.



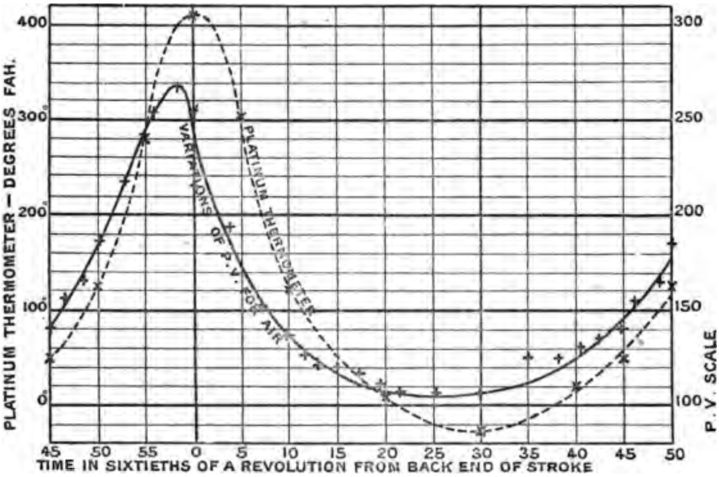
Curve showing agreement between Indicator and Platinum Thermometer.

Testing the Indicators.—The indicators were tested as nearly as possible under the conditions of the trials, and were carefully adjusted; they were daily oiled and cleaned, and tested for friction and back-lash. At the comparatively low speeds of the trials, it is hardly possible that they should have so considerable a lag as would be required to explain the difference between the indicator and the platinum thermometer.

Test of Sensitiveness.—A test of the sensitiveness of the platinum thermometer, in which the engine was run with air instead of steam in the cylinder, with a view to determine the probable amount of lag, is illustrated in *Fig. 19.* The lag could not have been greater than 2° with the temperature of the air rising at the rate of 100° per second, but it is not possible to obtain any form of indicator sufficiently sensitive and accurate to perform this test satisfactorily.

Superheating due to Wire-drawing.—*Fig. 20* gives an illustration of a different kind of steam-cycle, taken during the measurement of a valve leak, and showing a very unexpected amount of superheating. The temperature of the steam leaking into the cylinder under these conditions, as measured by the platinum thermometer on the piston, is shown by the dotted curve. The full curve below shows the temperatures deduced from the indicator on the assumption that the steam was saturated.

Fig. 19.



Temperature Cycle with Air in Cylinder.

Conclusions.—From the steam-cycle observed in the hole in the cover, it appears that, even within $\frac{1}{10}$ inch from the walls, the temperature of the steam is greatly affected by adiabatic compression and expansion, but that during comparatively quiescent periods of the cycle, such as the exhaust, the steam close to the walls is heated nearly to the wall temperature. In a single-acting non-condensing engine, with a moderate degree of compression, water is not likely to collect in the corners and recesses of the clearance; and the clearance contents, consisting chiefly of superheated steam, cannot be regarded as a primary cause of condensation of the admission steam. The fact that the platinum thermometer, after falling below the indicator at cut-off, crosses it again at a temperature slightly below that of the walls, and then rises considerably above it, shows that re-evaporation from the cover is probably complete some time before release, and that evaporation from a highly heated wall is probably a process of a rapid and explosive character. From the piston steam thermometer, the temperatures deduced for the indicated pressures seem to represent very fairly the average state of the main body of the steam, but the steam is probably slightly superheated during compression and admission, and slightly supersaturated during expansion and exhaust. The effects observed are probably too large to be explained by lag of the indicator. The superheating of the steam during admission may be partly explained by the further compression of the already superheated cushion steam, which in the present case formed one-fifth of the cylinder contents. It is also partly due to the kinetic energy of the inrush. In any case it is evidence that the steam supply from the boiler was fairly dry. Under the conditions of the trials it is impossible that the steam supply could have been superheated.

Fig. 20.

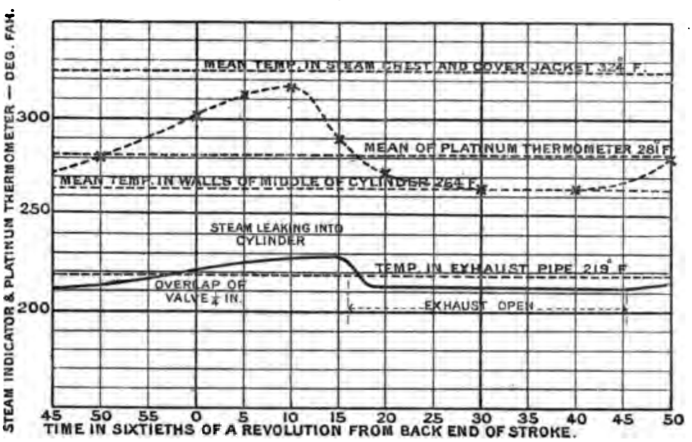


Diagram illustrating Superheating due to Wire-drawing.

thermometers in the steam-pipe and steam-chest indicated the normal temperature. The results of tests with a number of different thermometers showed the temperature of the steam during expansion, and during the early part of the exhaust, to be lower than that deduced from the indicator.

Dynamical Equilibrium of Expanding Steam.—Apart from the statical condition of equilibrium between steam and suspended water-drops, which depends on the size of the drops and on the value of the surface tension, there is also a dynamical condition in the case of rapidly expanding steam, which has not, so far as the Authors know, been previously noticed. When steam is expanding adiabatically, it requires, as is well known, a condensation equivalent to nearly one thermal unit per pound per degree Fahrenheit of fall, to keep it up to the saturation temperature. This condensation must take place chiefly on the surface of drops already formed. The temperature of the drops can be maintained only by continual evaporation. Unless the steam condensed is at a lower temperature than the drops, there can be no balance of condensation, and the temperature of the drop cannot fall. The lowering of steam-temperature required will evidently be proportional directly to the rate of condensation, and inversely to the surface exposed by the drops. Since the drops are at once foci of condensation and foci of heat, there must be powerful obstructive influences at work, and the lowering of temperature of the steam may therefore be considerable. In the absence of more certain indications these obstructive influences may be assumed similar in magnitude to those which limit the rate of condensation of steam on a metal surface. It is possible to make a numerical estimate of the order of the lowering of temperature. At n revolutions per minute, the initial rate of fall during expansion, in the experiments at one-fifth cut-off, may be taken as about $10n$ °F. per second, and the required balance of condensation as $8n$ thermal units per pound per second. If the wetness of the steam is 1 per cent., and the average diameter of the drops 0·000024 inch, the surface exposed per pound of steam would be nearly 480 square feet, and the lowering of the steam temperature at 100 revolutions would be about 2·7° F. The lowering required increases in direct proportion both to the speed and the diameter of the drops. With nearly dry steam at high speeds the initial lowering may be considerable, especially if the drops are large and few. If it be admitted that the temperature of rapidly expanding steam may fall considerably below its saturation temperature, it affords a possible explanation of a certain loss of efficiency in high-speed engines. From the behaviour of other vapours in a supersaturated condition, the pressure, if the steam remained dry, would be much more reduced for a given expansion of volume than it would be if the steam had time to maintain itself by condensation at its saturation temperature.

The statical condition mentioned affords an explanation of the unwillingness of steam to condense otherwise than on dust nuclei, or on drops of water already formed. It will be observed, however, that in order to account for a lowering of 2 or 3 degrees, the drops would have to be of so minute a size as to be invisible in the most powerful microscope. The linear dimensions of the drops actually occurring in steam-engine practice are probably between one hundred to one thousand times greater.

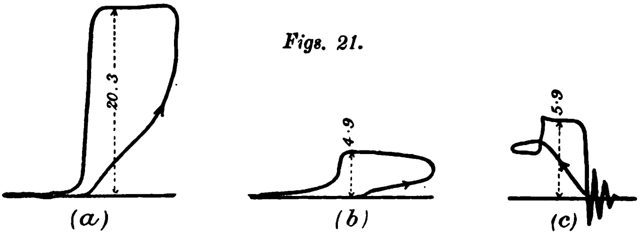
For the dynamical condition, on the contrary, the larger the drops the less the surface they expose, and the greater the fall of temperature required. A diameter of $\frac{1}{10,000}$ inch would mean a fall of about 50° F. below the saturation temperature at 400 revolutions per minute and wetness 1 per cent. In the absence of accurate knowledge of the properties of steam under these conditions, it is not possible to say exactly how much missing steam such a fall of temperature would account for. It would probably be between 5 per cent. and 10 per cent., according to the extent of the drop surface. The subsequent recovery of temperature, as more drops were formed and condensation proceeded, would simulate the effect of re-evaporation. The initial fall of temperature at a high speed, however considerable, would probably be of very short duration.

It is not improbable that steam in this supersaturated condition may tend to condense more readily on any surfaces exposed to it, which happen to be below the saturation temperature, than would be the case with ordinary wet saturated steam. The temperature of the steam itself may have some influence, as well as that of the surface on which it is deposited. For the same degree of adiabatic expansion, the heat abstracted from the walls would be the same, whether the steam condensed is dry and supersaturated, or wet and saturated, provided that in the latter case the suspended moisture is deposited along with the steam. The amount of condensation, however, might be greater in the case of the supersaturated steam, if the temperature of the steam itself has any influence.

Supposing the steam is many degrees below its saturation temperature during rapid expansion, no thermometer, however sensitive, could indicate the whole extent of the phenomenon. The rapid motion of the steam and the piston might tend to cool it, but the condensation on its surface would tend to keep it near the saturation temperature corresponding to the pressure. The lag of the thermometer would also make the reading too high. If, on the other hand, it is inconceivable that the thermometer should indicate anything but the saturation temperature, the differences observed must be due to a real difference of pressure, owing to the rapid vortical motion of the steam, between the centre and the circumference of the cylinder contents. Such differences must exist, and have often been regarded as important. It is probable that further experiments of this nature might throw some light on the question.

VALVE AND PISTON LEAKAGE.

The Slide Valve.—In estimating the amount of condensation in the cylinder by comparing the measured feed per revolution with the steam indicated by the diagrams, the valve and piston leak is generally assumed to be negligible. The effect of leakage, however, is in many ways so similar to that of condensation, that one may readily be mistaken for the other; and no estimate of condensation deduced from diagram and feed measurements can have any claim to consideration, unless the state of the valves and piston in respect of leakage is simultaneously investigated. In addition to trying the stationary test for leakage, which is very easily applied, the leakage was measured as accurately as possible under the actual conditions of running. The stationary test was found to be of little or no value.


Figs. 21.

Preliminary Leakage Tests.—To measure the leakage into and out of the cylinder, with the valve in motion under the conditions of running, but just not admitting steam directly, the piston was blocked and cards were taken, the barrel motion being obtained from the valve spindle, and the engine being driven by a motor. *Figs. 21, a* and *b*, are sample indicator-diagrams taken at the back end of the cylinder, in the above manner. The first, *a*, showing a maximum of 20·3 lbs., was taken on July 29th, diagram *b* on

August 29th, after the valve had been very carefully scraped and refitted; diagram *c* is one taken at the crank end on the same date.

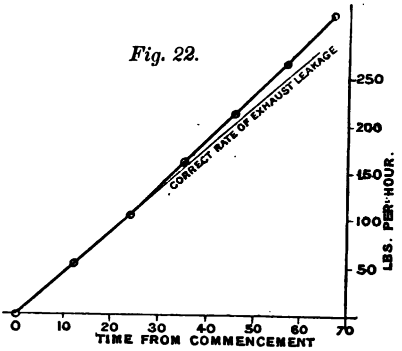
The following were the conditions of the test:—

Date.	Indicator-diagram (Type).	Average Pressure.		Maximum Temperature.	Volume of Cylinder.	Revolutions per Minute.	Leak.
		Gauge.	Cards, Maximum.				
				°	Feet.		Lbs. per Hour.
July 29 . .	*a*	97·0	19·2	301	0·360	41·8	38·6
August 29 .	*b*	81·0	4·87	316	0·648	75·7	30·6
August 29 .	*c*	81·0	5·3	..	0·063	75·7	3·6

A smaller indicated pressure in the second case was nearly compensated by the higher speed and greater volume, so that the resulting leak deduced is not very far from being proportional to the difference of pressure under which the leak took place. It would appear that the leakage is not merely a question of such minute differences of fit as those corrected in the scraping.

Direct Exhaust Leakage.—Preliminary trials showed the direct leakage of steam from the steam-chest into the exhaust to be by far the largest and most important. In order to measure this leakage as nearly as possible under the conditions of running, both the steam ports were blocked with lead, and the valve was driven by an electric motor, the piston being disconnected. The following are the results of two experiments made with the same valve setting as for the later series of trials at one-fifth cut-off. In the first trial, 112 lbs. were condensed in 25·17 minutes, at a gauge pressure of 91 lbs. per square inch, and the rate of leak appeared to increase slightly as the oil-film was gradually dissipated.

Fig. 22.



The effect of the dissipation of the oil-film is well illustrated in the second trial under the same conditions, for which the results of the separate weighings of the feed have been plotted in the curve shown in *Fig. 22.* In this trial 317·5 lbs. were condens‹

in 66·42 minutes, at a gauge pressure of 80·5 lbs. The initial rate of leak, however, which was taken as the correct value, agrees very closely with the previous trial.

To compare the results of different leakage trials, the leakage is provisionally assumed proportional to the difference of pressure. In the equation, leak in lbs. per hour $= k \times$ (difference of pressure); the constant k, which may be called the "rate of leak" of the valve, gives the leak per hour per pound difference of pressure. The rate of the direct exhaust leak for this particular setting of the valve is:—$k = 2 \cdot 98$. The round number $k = 3 \cdot 00$ for estimating the leakage corrections is evidently within the limits of error of the measurements. No trace of this leakage appears on the indicator-diagrams, which were of the most perfectly regular type throughout, as shown in *Fig. 8.*

Leakage of Unbalanced Slide-Valves.—With a view to investigate this question further, exactly similar tests were made on the smallest and largest valves of a quadruple-expansion engine. The H.H.P. valve gave a leak of 38 lbs. per hour, with a pressure difference of 100 lbs. between the steam-chest and the exhaust pipe. The L.P. valve gave 41 lbs., and 29 lbs. per hour, with pressure differences of 34 lbs. and 21 lbs. respectively. These valves are ordinary unbalanced slide-valves, with large bearing and guiding surfaces, and the fitting throughout is undoubtedly of a high order. The low-pressure valve was proved to be absolutely steam-tight when stationary.

Provisional Law of Leakage.—By applying the law of transpiration of liquids, assuming that the leakage takes place chiefly in the form of water, results were at once obtained which, considering the nature of the measurements, are remarkably consistent both for the balanced and for the unbalanced valves. The leakage of a liquid through a fissure of nearly uniform thickness, depending on the nature of the water- and oil-film, should be proportional directly to the difference of pressure and the perimeter of the port, and inversely to the width of the bearing surfaces. The latter factor is somewhat difficult to estimate for a moving valve, but for the present purpose the values given in the following Table are probably sufficiently approximate. If the fissure through which the leak takes place is of nearly the same thickness in each case, the rate of leakage k per lb. pressure per hour should be proportional to the perimeter p of the port, divided by the mean overlap l. Thus $k = \frac{Cp}{l}$, where C is a coefficient depending on the nature of the oil-film, which should be the same for the same type of valve. It might possibly be nearly the same for valves of different types, if it depends on some physical property of a mixture of oil and water.

TABLE V.—COMPARISON OF RATES OF LEAKAGE.

Valve considered.	Perimeter of Port, p.	Overlap (mean), l.	Ratio, $\frac{p}{l}$.	Observed Rate of Leak, k.	Deduced Value of $C = \frac{k\,l}{p}$.
	Inches.	Inches.			
Balanced, Robb	72	0·5	144	3·00	0·021
Unbalanced H.H.P. . . .	30	1·5	20	0·38	0·019
" L.P.	65	1·0	65	1·20	0·019
" "	65	1·0	65	1·38	0·021

The leakage coefficients observed with the different valves, are at least of the same order of magnitude. If so, we are justified in concluding that the leakage probably takes place in the form of water, and is proportional to the difference of pressure. It would also appear probable that such leakage is the normal state of things with a moving valve, and that the excessive leakage observed with the balanced valve, is not a defect peculiar to this type, but is simply a consequence of its comparatively large size. At the high speed for which this engine was designed, the leakage would not be a very serious matter. If the leakage were simply due to accidental tilting motions, or to bad fitting, it is difficult to see why the value of the coefficient C should be of the same order for valves of such different types. If, on the other hand, the leakage is not purely accidental, but follows a regular law, it is a matter of such practical importance as to be well worthy of further investigation.

Leakage into Cylinder after Cut-off.—From the experiments described on p. 31, it is possible to obtain an independent verification of the value of the coefficient C. With $l = \frac{1}{2}$ inch, and $p = 20$ inches, the values are $C = 0 \cdot 022$, and $C = 0 \cdot 020$ respectively. By means of the time integral of the expression $\frac{Cp\,(P_0 - P)}{l}$, the leakage taking place into the cylinder during expansion between the points 0·25 and 0·70 of the stroke, for trials XVI.–XX., amounts approximately to 6·0 lbs. per hour. The smallness of this result is due to the fact that the difference of pressure is small just after cut-off when the overlap is least.

Piston-Leak under Conditions of Running.—Under running conditions, the steam leaking past the piston was found to amount to 15·3 lbs. per hour under a mean pressure of 33 lbs. The piston-leak taking place during expansion between 0·25 and 0·70 of the

TABLE VI.— COMPARISON OF CARDS AND FEED WITH WALL-CYCLES.

Single-Acting Non-Condensing Trials. Cut-off at 0·200 of stroke.
Clearance, 10 per cent.; Release, 0·750; Expansions (cut-off to release) 2·83.

——	1	2	3	Means.	4	5	6	Means.	7
1. No. of trial . . .	XIX	XVIII	XX*a*	1, 2, 3	XVII*a*	XVI	XX*c*	4, 5, 6	XVII*b*
2. Duration in minutes	37·0	68·0	55·0	..	79·0	76·0	35·0	..	25·0
3. Mean revs. per minute	43·8	45·7	47·7	45·7	70·4	73·4	81·7	73·7	97·0
4. Mean gauge pressure	87·9	89·2	94·4	90·8	98·1	92·0	94·2	95·1	96·0
5. Gross feed per rev. .	·1422	·1437	·1483	..	·1094	·1036	·1000	..	·0856
6. Leakage correction .	·1004	·0976	·0990	..	·0697	·0627	·0576	..	·0494
7. Corrected feed per rev.	·0418	·0461	·0493	·0462	·0397	·0409	·0424	·0407	·0362
8. Calculated cushion steam	·0107	·0104	·0103	..	·0099	·0098	·0100	..	·0105
9. Total weight of fluid expanding in cylinder	·0525	·0565	·0596	·0567	·0496	·0507	·0524	·0505	·0467
10. Indicated weight at 0·250	·0407	·0414	·0437	..	·0418	·0394	·0408	..	·0393
11. Indicated weight at 0·700	·0466	·0456	·0488	..	·0460	·0436	·0454	..	·0426
12. Increase of indicated weight	·0059	·0042	·0051	..	·0042	·0042	·0046	..	·0033
13. Adiabatic condensation	·0019	·0020	·0021	..	·0020	·0019	·0020	..	·0019
14. Indicated evaporation	·0078	·0062	·0072	·0069	·0062	·0061	·0066	·0062	·0052
15. Calculated evaporation	·0076	·0073	·0070	·0073	·0048	·0046	·0041	·0046	·0035
16. Indicated condensation	·0118	·0151	·0159	·0146	·0078	·0113	·0116	·0099	·0074
17. Calculated condensation	·0148	·0142	·0136	·0142	·0092	·0089	·0080	·0088	·0067
18. Water present at 0·250 per cent. of feed (7) .	28·3	32·7	32·3	31·7	20·0	27·7	27·3	24·3	20·4
19 per cent. of fluid (9) .	22·5	26·8	26·7	25·8	15 7	22·3	22·1	19·6	15·9
20. Indicated HP. . . .	4·10	4·34	4·78	4·43	7·02	6·67	7·71	7·00	8·81
21. Lbs. per I.HP. hour	26·8	29·1	29·5	28·6	23·8	27·1	26·9	25·7	23·8
22. Condensation per I.HP. hour . .	7·6	9·5	9·5	9·1	4·8	7·5	7·3	6·3	4·8

stroke, under the conditions of trials XVI–XX, amounts to only 2·5 lbs. per hour, the greater part of the leak occurring during admission.

Comparison of Indicated and Calculated Condensation.—The results given in Table VI, with the exception of the first four and the last five lines, are expressed in terms of the weight of steam and water per cycle in lbs.

The differences of the weights in lines 9 and 10 are given in line 16. They depend entirely on the rate of leakage assumed for applying the very large leakage correction given in line 6. Considering the magnitude of this correction, it is most remarkable that the weights in line 16 should have so great a degree of consistency. A variation of only 3 per cent. in the rate of leak would be sufficient to explain the largest discrepancy from the mean. The order of consistency shown under such differences of speed and pressure, and probably also of lubrication, from trial to trial, is perhaps the strongest proof that could be given that the phenomenon of valve leakage is subject to regular laws, and deserves much more attention than it has hitherto received. Considering the range of speed covered by the trials, there is strong evidence that the rate of leakage is nearly independent of the speed of reciprocation of the valve. The evidence that it is simply proportional to the pressure is much less conclusive. Line 17 gives the total condensation at 0·250 of the stroke, calculated from the results of the observed wall-cycles for three low-speed trials. This assumes that the cyclical condensation per minute is independent of the speed provided that the temperature conditions remain unchanged. It is evident, on comparing the numbers in lines 16 and 17, that the indicated condensation agrees as closely as can be expected with this view. There is perhaps a slight indication of a greater rate of condensation at the higher speeds, but this is partly accounted for by the higher mean pressure and temperature corresponding to these trials. The results in line 17 were calculated to correspond to a gauge-pressure of 90 lbs.

The Nature and Effects of Valve Leakage.—The foregoing experiments would make it appear probable that a moving valve, however well fitted, is subject to a regular leakage of a peculiar type, which has not been previously suspected. The leakage appears to take place in the following manner. So long as the valve is stationary, the oil-film may suffice to make a perfectly tight joint, but as soon as it begins to move, the oil-film becomes broken up and partly dissipated. Water is being continually condensed on the colder parts of the surface exposed by the motion of the valve. This water works its way through, and breaks up the oil-film under the combined influence of the pressure and the motion. The continual re-evaporation taking place in the exhaust tends to keep the valve and the bearing-surfaces of the seat cool, and to maintain the leaking fluid in the state of water. The exhaust steam from the cylinder has the same tendency. The coefficients of viscosity of steam and water at the temperatures which occur in a steam-engine are not accurately known. But whereas that of steam increases with rise of temperature, that of water diminishes very rapidly. It is not improbable that the quantity of water which can leak through a given crack under a given difference of pressure, may be from twenty to fifty times greater than the quantity of steam which can leak under similar conditions. This agrees with well-known facts in regard to leakage, and explains how it is that the leakage in the form of water is so great. A few simple experiments were made with regard to the transpiration of water and steam under the conditions in question, and the leakage in the form of water was more than twenty times as great, the water being at a temperature below boiling point. The motion both of the water and the steam, owing to the high velocity, was certainly turbulent or eddying, which would have the effect of greatly increasing the resistance as compared with that due to viscosity, if the motion were steady. For the case of steady motion, comparative tests were made of the relative values of the viscosity of water, cold and hot. The measurements were not sufficiently accurate to give the law of the variation of the viscosity with temperature above 212°; but it appeared that the viscosity at 212° F. was only one quarter of that at 62° F., and that it continued to diminish very rapidly. Under the actual conditions of the valve-leak experiments, the water leak is more likely to have been between forty or fifty times the steam leak. An explanation is thus furnished of a possible form of leakage, indirectly due to condensation and re-evaporation—so many times greater than the steam leakage, which, alone, engineers have been in the habit of contemplating, that it might well claim attention on its own merits, apart from the very limited number of valves on which it has hitherto been possible to make direct experiments.

The analysis of a large number of observations, in addition to the few made by the Authors, leads to the conclusion that all valves leak more or less when in motion, and that in many cases the greater part of the missing quantity is to be attributed to leakage of this description. Whatever the precise manner in which the leak takes place, it appears to be nearly proportional to the difference of pressure, and to be in most cases independent of the speed. In any case, it appears probable that the leakage is connected in some way with the condensation taking place on the valve surfaces. If so, it may evidently be greatly reduced, if not entirely cured, by jacketing, or otherwise heating the valve-seat, to minimise the condensation.

These views have an important bearing on the design of valves. For low-speed engines, separate steam- and exhaust-valves should possess advantages over the ordinary slide-valve. The superiority of the compound engine would also appear to be partly due to the great reduction of possible leakage.

GENERAL CONCLUSIONS.

On the Rate of Condensation of Steam.—The most important general conclusion to be derived from the experiments is that the rate of condensation of steam on a metallic surface is limited, and is proportional to the difference of temperature. The small increase observed in the ratio $\frac{Q}{A}$, p. 21, with increase of speed, may be most naturally explained as due to the diminished thickness of the water-film deposited, and the smaller range of the metal cycle. The slightly smaller values observed on the cover as compared with the sides, though possibly due to slight differences in soldering or in the form of the cycle, may also be attributed to the state of the surface. Taking these factors into consideration, the probable rate of condensation of steam on a clean and dry metal surface is found to be 0·74 thermal units per second per square foot per 1° F. difference of temperature at 300° F. Expressed in more familiar quantities, the rate above given would correspond in a surface condenser to the condensation of 27 lbs. of dry steam per square foot per hour for a difference of temperature of 10° F. between the steam and the surface.

Condensation in Terms of Temperature Distribution.—Assuming that the amount of condensation is limited by the rate of condensation of steam given by the above law, the problem of estimating the amount of condensation taking place in any given engine with any given steam cycle, is reduced to the (comparatively) simple problem of determining the temperature distribution on the cylinder walls while the engine is running. Given the temperature distribution, the condensation is inferred by measuring the condensation areas on the cycle diagram.

Limit of Cyclical Condensation for any given Cycle.—That it should be necessary to observe the temperature distribution in any case, in order to be able to deduce the cyclical condensation, may appear at first sight a somewhat disappointing result. The form, however, of the law of condensation and re-evaporation as deduced from the experiments, leads directly to a limiting value of the cyclical condensation, and gives a result of striking simplicity, which is undoubtedly applicable to a large number of important cases. If the conditions of external and internal heat loss are supposed to be such that the mean temperature of the clearance surface, on which the greater part of the initial condensation takes place, is reduced to the mean of the steam cycle, it is plain that the condensation and evaporation areas on the cycle diagram will be equal. If the temperature of the clearance surface falls below this point, evaporation will be incomplete. Water will then accumulate in the cylinder until a balance is attained by the mechanical removal of the excess. It is obvious that all steam condensed on the surface, and then mechanically rejected in the form of water, represents the communication of a quantity of heat to the walls, equivalent to the total heat of the steam condensed, and thus rejected, reckoned from the temperature at which the water is thrown off from the walls. The quantity of heat thus communicated per lb. of water rejected, may be between twenty and fifty times greater than that communicated by the condensation and subsequent re-evaporation of an equal quantity of steam.

If the engine starts cold, and the surfaces are gradually heated by the action of the steam, it is clear, from the same considerations, that the rise of temperature up to this point, so long as water is being mechanically rejected, will be extremely rapid. The mean of the steam cycle is, therefore, on the provisional law, a natural minimum of temperature for the wall surface, corresponding to a maximum limit of condensation for any given cycle.

In order to deduce the limiting value of the condensation per square foot per hour for any given cycle, it is simply necessary to draw the cycle diagram corresponding to the indicator-diagram, and to rule across it the line representing the time average of the steam temperature, as shown, for instance, in *Fig. 18.* The area above this line is the maximum condensation area corresponding to this particular cycle. The maximum value of the condensation, measured in thermal units per hour, is forty-five times this area measured in degrees F. and sixtieths of a cycle. As a general rule, the condensation must be less than this limit, because the temperature range of the surface of the metal, which is far from negligible at low speeds, has the effect of diminishing the condensation area, and also the available evaporation area to a nearly equal extent, although it does not materially affect the condition that the mean temperature of the wall surface should be the same as that of the steam-cycle.

If t' is the mean temperature of condensation, and t'' that of re-evaporation, and L′, L″, the corresponding heats of vaporization, $L'' - L' = 0 \cdot 70 \ (t' - t'')$, approximately. The quantity of heat Q given up to the walls by condensation and re-evaporation per pound of steam, is given by the formula, $Q = 0 \cdot 30 \ (t' - t'')$. If t° is the mean temperature of the walls, the quantity of heat given up to the walls by each pound of steam condensed at t' and rejected without re-evaporation, is $L' + t' - t^\circ$. If the range $t' - t''$ is 50° F., this quantity is more than fifty times as great as Q. The mean temperatures of condensation and evaporation, t' and t'', are found by a process analogous to that of finding the centres of gravity of the corresponding areas. Each temperature is weighted in proportion to its difference from that of the wall surface, which determines the rate of condensation at each point of the cycle. Following the usual notation for the centre of gravity formula,

$$t' = \frac{\Sigma\, t\, (t - t^\circ)}{\Sigma\, (t - t^\circ)}$$

where t, t°, are the temperatures of the steam and walls at any point of the cycle. The quantity of heat given to the walls by the condensation and re-evaporation of dry steam, is given by the formula, $Q = 0 \cdot 30 \ (t' - t'')$, per pound. A more elaborate or apparently exact formula, is useless, because the value of the constant 0·30 in this expression, being the value of the change in the total heat of steam per 1°, is one of the most uncertain elements in the whole theory of the steam-engine. According to the experiments of Regnault and Griffiths, the mean value of this constant between 30° and 100° C. should be 0·40, but it appears not improbable that its value diminishes with rise of temperature.

It has often been pointed out that, as a result of the comparatively small rate of increase of the total heat of steam with rise of temperature, a relatively small loss of heat, or a slight change of conditions, is competent to account for a considerable change in the initial condensation. The balance is extremely delicate, and is very easily turned. The further possibility, that there should be a limit of condensation, results from the form of the law of condensation, and could not have been foreseen so lon as the rate of condensation was regarded as infinite. When this limit is reached, the conditions as regards increase of condensation are extremely stable. If the limiting range $t' - t''$ is 50° F., which is not uncommon in compound engines, a loss of 15 T.U. is sufficient to account for each pound of initial condensation and re-evaporation at this range. But if at this point the rate of loss of heat is suddenly doubled, the initial condensation will be increased by less than 2 per cent. If, on the other hand, the rate of loss of heat were reduced to one-quarter, the initial condensation and the range of temperature between condensation and re-evaporation would each be reduced to one-half of their limiting values. It is also interesting to observe that the form of the law of condensation would make the limiting value of the condensation in any cylinder depend chiefly on the temperature range in that cylinder. Of all the results which have been empirically established with regard to cylinder condensation, this result has always been regarded as the most certain. To express the result more accurately, according to this law, the limiting value of the condensation, when measured in pounds per hour, should vary as the area included between the steam-temperature cycle curve and the line representing the mean temperature of the steam-cycle. The limiting value should also increase slightly with increase of speed, because the temperature range of the metal surface is reduced, and the effective condensation area is thereby increased.

Correction for the Metal Surface Cycle.—It is clear that no simple formula can be constructed to take account of all the possible varieties of cycle. The correction is not large, and might be neglected if it were not that it varies with the speed and with the point of the cycle considered. In the majority of cases which occur in practice, the maximum point of the wall-surface temperature cycle is found to coincide very nearly with the point of cut-off, and the point at which the steam-cycle curve crosses the wall-surface curve is generally very near the point of release. For these two points of the cycle, in the case of limiting condensation, a sufficiently approximate correction can be applied for the effects of variation of speed by the following simple method. The condensation area is measured from the line of mean wall-temperature, and up to the line of cut-off or release as required. The area so measured is then reduced in the proportion $\frac{(1 + \sqrt{n})}{(3 + \sqrt{n})}$ for the point of cut-off, and in the proportion $\frac{\sqrt{n}}{(3 + \sqrt{n})}$ for release. The correction at the point of cut-off varies from 25 per cent. at n = twenty-five revolutions per minute to 9 per cent. at four hundred revolutions per minute, if the steam-cycle remains the same. This formula assumes a cast-iron cylinder. The main effect of making the correction is to increase the value of the condensation constant from 0·61 T.U. to 0·74 T.U. per degree-second per square foot, as deduced from Table III. It also has the effect of making the observations agree rather better with the calculations at higher speeds, both in Tables, pp. 19 and 34.

Effect of Initial Wetness of the Steam.—In view of the observations on the effect of the condensation of wet steam, it is interesting to make an estimate of the possible increase of condensation thereby produced. It is, however, necessary to make a few assumptions, which are probably not in all cases justifiable.

Let x be the dryness fraction of the steam condensed at a temperature t', and let it be assumed that the proportion of suspended moisture $(1 - x)$ is all deposited on the walls together with the condensed steam. Let it be further assumed that the whole of this water is re-evaporated non-explosively at an average temperature t''. The mean temperature of the wall-surface, t°, will be somewhere between these two extremes, and will be modified by conduction from the neighbouring parts of the cylinder. To simplify the conditions the point considered may be situated on the clearance surface near the middle of the cover, or the cylinder to be so large and thin that the effect of conduction may be neglected. The other losses may also be supposed negligible in comparison with that due to re-evaporation.

If L' L'' are the latent heats of vaporization at the temperatures t' t'', $L'' - L' = 0{\cdot}70\,(t' - t'')$. The heat supplied by condensation per pound of steam condensed is L'. The weight of water deposited by the condensation of 1 pound of steam is by hypothesis $\frac{1}{x}$, and supplies heat to the amount $\frac{(t' - t'')}{x}$ in cooling from t' to t''. The heat abstracted by re-evaporation of this water is $\frac{L''}{x}$. The balance of heat abstracted is therefore $\frac{L'' - t' + t''}{x} - L'$, which may be written in the form $\frac{L'(1 - x)}{x} - \frac{0{\cdot}30\,(t' - t'')}{x}$. Unless the temperature of the wall is maintained by external agency, it will therefore continue to fall until $(t' - t'') = \frac{L'(1 - x)}{0{\cdot}30}$. For instance, the effect of 5 per cent. wetness is to lower the wall temperature until the range $t' - t''$ is 150° F.

Effect of Water in the Cylinder.—After the accumulation of water in the cylinder has commenced, it is not quite so clear whether the effect of the water would be to increase or diminish the total condensation. It appears probable there would not be a great change. In a working cylinder, the water could not accumulate to any considerable thickness, except in special pockets. If the water were present in sufficient quantity to be thrown into spray, and thoroughly mixed up with the steam, so as to expose a large surface to its action, the water so broken up would almost certainly be carried out of the cylinder with the steam, in proportion to the minuteness of its subdivision. The film left on the surface would, therefore, probably be very thin, and would not seriously affect the result, either in the direction of increase or diminution. To test this supposition, the law of condensation was applied to calculate the mean wall temperatures and the amounts of condensation observed by Donkin in his "Revealer" experiments, in those cases in which water was probably present. The results thus calculated agreed with the observations within the limits of error of the measurements.

Case of Limiting Condensation.—It would appear from the above considerations that the case of limiting condensation in which the re-evaporation is incomplete requires to be treated separately, not only on account of its superior simplicity, but also because the possible variation of the condensation under these conditions differs so greatly from the case in which re-evaporation is complete. In order to test whether any given cylinder is actually in this condition, it is only necessary to insert a thermometer in some convenient hole in the metal of the clearance surface, and to compare the temperature indicated with the mean of the steam cycle. If the clearance surface is found to be at or near the critical temperature, the limiting value of the condensation has probably been reached, and can be approximately calculated by the above method. Next to the average indicator diagram of the trial, the most important datum required for the application of the method is the extent of the clearance surface. In cases where this is not known, and the diagrams are not given, but only the initial and exhaust pressures, a rough estimate of the probable limiting condensation in lbs. per hour, double-acting, may be formed by multiplying the temperature range by four times the area of the piston-face in square feet.

The high-pressure cylinder in Table VII was jacketed with its own exhaust steam. This fact, however, would hardly be enough to account for the missing quantity being nearly twice as

TABLE VII.—ILLUSTRATION OF LIMITING CONDENSATION.

Trial of "Ville de Douvres."[1]

Compound, 2,977 I.HP.; Feed per hour, 61,800 lbs.;
Revolutions per minute, 36·8.

1. Cylinder considered	High Pressure.	Low Pressure.
2. Point of stroke	Cut-off	Release.
3. Percentage of feed missing	19·4	27·5
4. Missing quantity . . . lbs. per hour	12,000	17,000
5. Mean of steam temperature cycle . . . ° F.	287	205
6. Range " " " . . . "	86	81
7. Range of pressure . . . lbs. per square inch	80	24
8. Clearance surface . . . square feet	85	211
9. Condensation area . . . ° seconds	794	804
10. " corrected for metal cycle	625	540
11. Thermal units absorbed per square foot per minute	460	400
12. Clearance condensation . . . lbs. per hour	5,200	10,500
13. Barrel surface exposed . . . square feet	47	140
14. Mean condensation area, corrected	230	200
15. Barrel condensation . . . lbs. per hour	1,100	2,500
16. Total " . . . " " "	6,300	13,000
17. Remainder to be accounted for	5,700	4,000

great as the condensation limit. Leakage of the high-pressure valves or piston appears probable. The remainder, line 17, in the case of the low-pressure cylinder, corresponds fairly well with the probable wetness of the steam due to adiabatic expansion combined with partial drying due to friction, re-evaporation and other causes. The examination of a number of such cases leads to the following general conclusions. In the high-pressure cylinder, it is not generally likely, owing to the probable dryness of the initial steam, and also to the greater probability of partially explosive evaporation, that the condensation should reach its limiting value. Nevertheless, in the great majority of cases, the missing quantity at cut-off in the high-pressure cylinder is much greater, often many times greater, than can be accounted for on the supposition of limiting condensation, according to this method of analysis. To account for this, the condensation must either be proportional to the density of the steam, or the greater part of the missing quantity represents leakage. The former supposition conflicts with the Authors' experiments and those of Donkin, but is in agreement with those of some other observers. The latter supposition would generally require rates of valve and piston leakage similar to those observed, and would appear the most natural explanation in the light of these experiments.

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[1] Proceedings of the Institution of Mechanical Engineers, 1892, p. 158.

In the low-pressure cylinder of an unjacketed engine, the condensation may frequently have its limiting value, owing to the initial wetness of the steam. It is also much more important than the leakage for two reasons. The condensation is greater, because the initial surface exposed is much larger. The leakage is less, because the difference of pressure on the valves and piston is much smaller. It is also evident that the perimeter of the ports and piston, upon which the leakage mainly depends, varies directly as the linear dimensions, whereas the surface exposed for condensation varies as the square of the diameter.

Case of Partial Condensation or Complete Re-evaporation.—It is evident, from the smallness of the results obtained in the small single-acting engine at low speeds, and from many similar results obtained by other observers, that, in the case of the simple engine, when working at a moderate ratio of expansion, the initial condensation is often very far below its limiting value. Re-evaporation is probably complete on the clearance surfaces, either at release, or at a very early period in the exhaust, and the walls are probably dry for most of the return stroke. The case of complete re-evaporation, or of partial condensation, as it may be called to distinguish it from the special case of limiting condensation, does not admit of the same simplicity of treatment as the limiting case. The temperature conditions are evidently far less stable, and the amount of cyclical condensation, which depends on the balance of heat loss and supply, is liable to be affected to a much greater extent by the variations of the conditions of running, and by differences of type and arrangement in different engines. It would, therefore, be unsafe to attempt to apply the results deduced from any one engine, under special conditions, to any other engine. At the same time it is possible that some light may be thrown on a very complicated problem by the careful consideration of the results observed in a particular case. The amount of cyclical condensation would appear not to be greatly affected by a moderate variation of speed. Some increase is to be expected, both on account of the diminished range of the metal cycle and also on account of the greater convective action of the exhaust steam; but the former cause partly tends to compensate itself by producing a higher wall-temperature, and the latter depends so much on the initial pressure and on other conditions that it cannot be satisfactorily represented by a formula.

Effect of Varying the Conditions of Running.—Assuming as a first approximation that the cyclical condensation is independent of the speed, it may most conveniently be expressed either in thermal units per minute or in pounds of water per hour.

(*a*) *Variation of Cut-off.*—From observations made at one-half, one-third, and one-fifth cut-off, it is inferred that, when the engine is working single-acting, non-condensing, the cyclical condensation measured per minute at cut-off is to a first approximation independent of the ratio of expansion. If the condensation is measured as a percentage of the indicated steam at cut-off (excluding cushion steam), this result is equivalent to the statement that the percentage condensed increases nearly in direct proportion to the ratio of expansion, defined as being the ratio of the volume occupied by the feed steam at cut-off to the volume of the cylinder. It is not convenient to measure the condensation as a percentage, either of the whole cylinder contents or of the whole cylinder feed, because these involve cushion steam and leakage. In a double-acting engine, the condensation on the barrel surface is necessarily less, and the temperature of the clearance surfaces higher and less variable. The change of each term would therefore be less, and it would seem probable that a similar compensation would occur, leaving the condensation at cut-off unaltered by change in the ratio of expansion. The formula is very attractive on account of its simplicity, which is the first desideratum in a formula of this kind intended to cover roughly a variety of conditions. Its applicability to the case of the double-acting engine is not to be suggested were it not that it appears to represent very fairly many of the most reliable results. In a large class of engine trials, the effect of varying the ratio of expansion r, on the observed percentage z of the cylinder feed condensed at cut-off, is closely represented by the semi-empirical formula of Thurston, $z = a \sqrt{r}$, where a is a constant depending on the other conditions. The numerical value of a for the engine would be 15. According to the Authors' formula $z = \frac{100\, c\, r}{100 + c\, r}$, where c has the numerical value 10 in the present case. It is remarkable that these two formulas, which are at first sight so totally dissimilar, should give results not differing by more than 2·5 per cent. throughout the whole range, from three to twenty expansions. From three to one expansions, the Authors' formula would appear to be preferable, as that of Thurston generally gives results which are too high as compared with experiment.

(*b*) *Double- versus Single-Acting.*—It is possible to draw general conclusions from a consideration of the effect on the distribution of temperature. In a double-acting trial there would be practically no effect of convection of heat by the piston. The gradient of longitudinal conduction would be halved at an early cut-off, and would practically disappear at a late cut-off. The effect would be to raise the temperature of the barrel portion of the admission surface very materially. The condensation on the piston would also be reduced, probably to less than one-half. Against these reductions the effect of the piston-rod at the crank end, and of conduction of heat to the framework of the engine, are to be set, which would tend to lower the temperature at that end as compared with the back end of the cylinder. It is evident that the condensation reckoned as a percentage of the steam would be considerably reduced. For the engine under review, the reduction is estimated at 30 per cent.; that is to say, the total condensation per minute, instead of being doubled, would be increased by about 40 per cent. as an outside estimate. In the case of large engines, the effect of conduction being negligible, the percentage saving by double action would be less.

(c) *Variation of Initial Pressure.*—The initial pressure was not sufficiently varied to give any direct information on this point, but experiments show indirectly that the effect is much more complicated than might be supposed. The external loss of heat from the cylinder would be increased nearly in proportion to the increase in the difference of temperature from the surroundings. The internal loss by re-evaporation and by the exhaust steam might be modified in a very different way. The experiments would appear to indicate, as has been suggested by Kirsch, that re-evaporation from the more highly-heated portions of the walls is of an explosive character; that is to say, that a portion of the water-film is blown off the walls without abstracting a full equivalent of its latent heat of vaporization. Condensing at atmospheric pressure with an initial steam temperature of 330° F. to 325° F., the temperature of the cover was over 300° F., and the platinum thermometer in the cover appeared to show that re-evaporation was complete almost as soon as the indicated temperature fell below this point. This observation at once suggested the partially explosive character of the evaporation as a possible explanation of the high temperature attained by the cover. The same explanation probably applies to a less extent to the hotter parts of the barrel surface, which appeared to have been gaining much more heat by condensation than they were losing by re-evaporation. On the cooler parts the balance of heat supplied would be simply that due to the small difference in total heat between the steam condensed at a higher and evaporated at a lower temperature. At one-fifth cut-off the evaporation apparently ceased to be explosive at a temperature between 270° F. and 260° F. The steepness of the temperature gradient along the sides of the cylinder cannot otherwise be satisfactorily accounted for. If the possibility of explosive evaporation at higher temperatures is admitted, depending partly on the diminished surface tension of the water and partly on the greater density of the steam, it is clear that the condensation may not necessarily increase continuously with increase of initial pressure. This result was arrived at independently of Kirsch, from the evidence of the observations. Kirsch makes the suggestion, not from direct experiment, but as a possible explanation of the smallness of the condensation observed in practice as compared with that which would theoretically be required, supposing that the surface of the walls were raised to the temperature of the steam, on the usual assumption that the rate of condensation of steam is practically infinite. Without further evidence, it would not be fair to conclude that re-evaporation at higher temperatures is always of this character. The conditions of the hole in the cover are not quite the same as those of the plane surface; but the observations suggest a possibility which obviously requires consideration. On one occasion a curious effect was accidentally obtained as an illustration of the possible consequences of explosive re-evaporation. When the electric-lighting engine was unexpectedly shut off, the boiler-pressure rapidly rose nearly 10 lbs. above the usual limit. This produced a rise of temperature of 6° F. at the back end of the cylinder, but the temperature at the middle of the cylinder rose more than 15° F., from 264° F. to nearly 280° F. The other conditions of running were unchanged with the exception of a slight increase of speed. As a general rule the changes of temperature at this point of the side were less than those at the back end of the cylinder. It is difficult to account for this abnormal rise of temperature, except by supposing that the re-evaporation at this point ceased to be normal, and became partially explosive. It will be observed that the change of temperature took place at that part of the scale which for other reasons appears to be the critical point in a non-condensing engine.

(*d*) *Variation of Wetness.*—It has long been recognized that the presence of water in the cylinder or of priming in the steam, must have the effect of increasing condensation. In a small single-acting engine, at low speeds, and without special precautions as to lagging the cylinder or drying the steam, practically conclusive evidence was obtained that the action of the metal alone was competent to produce the observed effects. At the same time an illustration was furnished of the abstraction of heat by the condensation and re-evaporation of wet steam, which shows that initial wetness of the steam is probably one of the most powerful factors in increasing cylinder condensation.. Provided that the initial wetness is small, and that the lowering of the wall-temperature produced by it is not sufficient to greatly change the other conditions upon which the balance of heat depends, it is possible to represent the effect in a simple manner by combining the formula already given with the law of condensation. If $t' - t^{\circ}$ be the mean difference of temperature between the steam and the walls during condensation when the steam is dry, the balance of heat supplied by condensation and evaporation is approximately $0 \cdot 60\,(t' - t^{\circ})\,W'$ per hour, where W' is the weight condensed. For a small lowering of wall-temperature, the balance of heat required would not be greatly reduced, and the condensation would be increased nearly in the same proportion as the difference of temperature $t' - t^{\circ}$. Upon these assumptions, it is found that, for initial steam of a percentage dryness $100\,x$, the initial condensation, as compared with that due to dry steam, is increased by a percentage given by the formula $\frac{100\,L\,(1 - x)}{1 \cdot 2\,(t' - t^{\circ})}$. If the temperature difference for dry steam is 30 F°. and $L = 900$, the effect of 1 per cent. of wetness would be to increase the initial condensation by 25 per cent. In the majority of partial condensation cycles, the errors involved in the above assumptions are of such a nature as to make this formula hold through a somewhat wider range than would otherwise be the case, but it cannot be trusted beyond 50 per cent. increase, and should be regarded, in any case, as showing rather the general nature of the effect than its absolute magnitude.

(*e*) *Variation of Back Pressure.*—For a given initial pressure, the wetness of the exhaust steam, due to adiabatic expansion, will depend on the back pressure. The cooling of the internal surfaces during exhaust, apart from re-evaporation, will depend on the wetness of the exhaust steam quite as much as on its temperature. From the reports of trials in which the jacketed surfaces of cylinders, valve-chests, and receivers are given, it is possible to estimate that the rate of abstraction of heat by wet steam in motion under such conditions does not probably ever exceed 1 T.U. per square foot per minute per 1° F. difference of temperature, and may be very considerably less. If the wetness per cubic foot, and not per pound, of the steam, is considered, it would appear probable that the cooling effect of the exhaust steam in a condensing engine may often be actually less than in a non-condensing, but that on the average there is no decided difference.

(*f*) *Effect of Compression.*—If there were no interchange of heat between the walls and the steam, compression to the initial pressure would restore the cushion steam to its initial state. The volume at compression in the present case was one-third of the volume of the cylinder and clearance, and probably included at least two-thirds of the heat abstracted by the exhaust steam. This would explain the very considerable superheating of the steam observed during compression close to the walls. The effect of an early compression may thus be regarded as equivalent to a considerable reduction of the loss of heat due to the exhaust steam, which is probably, next to initial wetness, the most potent factor in abstracting heat from the walls. The effect of an early release is probably similar to that of an early compression in reducing condensation, though it acts in a different way. In the case of partial condensation an early release allows less time for the condensation of wet steam on the colder parts of the walls towards the end of the stroke, and more time for the walls to dry before the return stroke of the piston, a condition probably unfavourable to piston leakage. It is possible that an early release may diminish the condensation and the exhaust waste sufficiently to more than compensate for the loss of area on the indicator diagram. In the cases of limiting condensation, on the contrary, the effect of an early release may be to increase the condensation, and to lower the temperature of the walls by increasing the available evaporation and condensation areas.

Effect of Superheating.—It would appear improbable that superheated steam can supply much heat to walls which are below the saturation temperature, except in so far as it condenses on the walls. Since the superheat is a very small proportion of the total heat, it may be naturally supposed that the rate of condensation of superheated steam is not very different from that of saturated steam. If the superheat be $s°$, and the saturation temperature of the steam t', the quantity of heat supplied to the walls by the condensation and re-evaporation of a weight, W$'$, of superheated steam will be $Q = 0 \cdot 60\ (t' - t°) + 0 \cdot 5\ s$ per pound, where $t°$ is the mean wall-temperature, and the specific heat of steam is taken as $0 \cdot 5\ s$. The law of condensation, making the same assumptions as in the case of initial wetness, leads to a similar formula for the reduction of the initial condensation by a small degree of superheating. If $(t' - t^\circ)$ represents, as before, the difference of temperature between the steam and the wall-surface when the engine is using saturated steam with the same cycle, the percentage reduction of the initial condensation, effected by the use of slightly superheated steam, is given by the expression $\frac{100\,s}{2 \cdot 4\,(t' - t^\circ)}$. This formula may be taken as holding approximately for the clearance surface, provided that s is not greater than $\frac{1}{2}\,(t' - t^\circ)$. The reduction of condensation in the case in which $s = t' - t^\circ$ amounts to about 33 per cent. instead of 41 per cent. as given by the above formula. The difference of temperature between the steam and the clearance surface is diminished approximately in the same proportion.

Besides diminishing the initial condensation by raising the temperature of the wall-surface, the use of superheated steam diminishes the wetness during expansion, and therefore considerably reduces the exhaust waste and the abstraction of heat by the condensation of wet steam towards the end of the stroke. It is also probable that it may tend to diminish leakage.

Effect of Jacketing.—The effect of jacketing a cylinder with steam at boiler-pressure, is to raise the temperature of the jacketed walls very nearly to that of the boiler if the jackets are working properly. According to the law here proposed, the condensation on the jacketed surfaces would be practically negligible, and the clearance surfaces are by far the most important. In large engines it would consequently be of little use to jacket the sides, but in small engines the clearance surfaces would also be heated by conduction so as to be practically jacketed. It is not improbable that a part of the economy due to jacketing, especially in small engines, is owing to the reduction of leakage. The valves and valve-seats become so heated by conduction that the possible water-leakage is minimised. From the same point of view, the drying of the steam in jacketed receivers must have a beneficial tendency, as there is then less water available to cool the valve-surfaces by re-evaporation in the exhaust.

Variation of Size and Surface.—The effect of variation of surface exposed, and particularly of the extent of the clearance surface, is probably different according as the condensation is of the partial or limiting type. If the main factor in the abstraction of heat from the clearance surface is the condensation of wet initial steam, as is probably the case when the condensation has its limiting value, the amount of heat abstracted and the initial condensation will vary simply as the surface exposed. In this case it is of primary importance to know the full extent of the clearance surface on which the greater part of the condensation takes place. In order to reduce the loss as far as possible, the extent of the clearance surface should be reduced to a minimum. If, on the other hand, the initial steam is dry, and the condensation is of the partial type, the extent of the clearance surface is a matter of much less consequence, because increase of surface has the effect of raising the temperature. It is probably best, in the case of partial condensation, to neglect differences of clearance surface in comparing different engines, and to take the clearance surface at each end of the cylinder as being πd^2, where d is the diameter. The actual clearance surface cannot be more than 50 per cent. less than this, and is seldom more than 50 per cent. greater. For all practical purposes the equivalent clearance surface forms a sufficient basis of comparison. But the barrel surface exposed up to cut-off, allowing for the time of exposure, may be very simply represented, if desired, by the addition of the term $l\,d\,c$, where c is the cut-off fraction, and l the stroke. The state of the surfaces is not important if the rate of condensation of steam is regarded as the main factor in limiting the amount of heat absorbed. The presence of a thin film of grease or rust may make the cycles observed at a given depth in the metal of smaller range, but will not really make much difference in the amount of condensation, unless the film is so thick and obstructive as to greatly increase the surface range of temperature, which is probably seldom the case.

Effect of Conduction.—In two similar cylinders of different linear dimensions, but with the same distribution of temperature, the loss of heat from the admission surface by conduction, will be proportional to the thickness of the metal. The initial loss due to conduction, reckoned as a percentage of the steam, for cylinders of the same thickness, will vary inversely as the cube of the linear dimensions. It is necessary to exercise caution in applying the results deduced from small engines to large. In order to estimate the probable effect of conduction, the temperatures have been compared with those obtained by Donkin with a cylinder 6 inches in diameter, and 8 inches stroke. The Authors conclude that the effects of conduction in their engine are not to be neglected as compared with larger machines, but probably do not amount to more than five or ten per cent., and are not such as to seriously vitiate the general nature of their conclusions. In cylinders of different shapes, under similar conditions of running, the loss due to conduction at cut-off reckoned as a percentage of the indicated steam, would vary as $\frac{t}{d\,l^2}$, where t is the thickness of the metal, d the diameter, and l the length of the cylinder. For instance, the effect of conduction in the cylinder above mentioned, would be nearly five times as great as in that of the Robb engine.

Formulæ of Condensation and Leakage.—From the foregoing considerations it will be evident that, when the required data are available, or when it is possible to observe the temperature distribution, no formula can be regarded as being at all satisfactory. It is, nevertheless, convenient to have a simple approximate formula for the purpose of making rough estimates and comparisons, and also as exhibiting the results of the investigation in a brief and compact form. In the case of limiting condensation, when neither the cards nor the extent of the clearance surface are given, the limiting value of the condensation W′ at cut-off, expressed in pounds per hour, may be estimated from the formula,

$$W' = \pi\, d^2 \,(t' - t''),$$

where d is measured in feet and t in degrees Fahr. This formula assumes that the clearance surface is $\pi\, d^2$, but makes an allowance of 20 per cent. for the barrel surface. It also assumes that the cut-off is at or near mid-stroke, and that the drop of pressure during admission is small. The factor $(t' - t'')$ is supposed to represent the total range of the steam. The latent heat of the steam is taken as 900. The result, thus estimated, may be corrected by substituting the proper value of the latent heat in each case, and may then be reduced in the proportion $\frac{(1 + \sqrt{n})}{(3 + \sqrt{n})}$, to allow for the effect of the probable range of the metal cycle.

The case of partial condensation may be roughly represented by the formula,

$$W' = C \times S = S\,(t' - t^{\circ}),$$

where C is the condensation in pounds per hour at cut-off per square foot of total equivalent clearance surface S of the cylinder, supposed unjacketed. The condensation factor C is a function of the initial and exhaust temperatures and of the external conditions, but may be taken as being approximately independent of the speed and the ratio of expansion. Apart from superheating or jacketing, the value of C is probably most affected by the degree of compression. The condensation factor C may also be interpreted as the mean difference of temperature $(t' - t^{\circ})$, between the walls and the admission steam, reduced to one-half cut-off.

The effect of size and surface, and of double or single action, may be supposed included in the expression for the surface S. This factor should be taken as $2 \pi d^2$ double-acting, and as πd^2 single-acting.

The effect of jacketing in large engines may be represented by simply omitting the jacketed area from the factor S; but the effect of conduction in small engines cannot be satisfactorily included in the formula. If W is the weight of feed in pounds per hour, accounted for by the indicator at cut-off, and W° the total missing quantity per hour, $W + W^\circ$ represents the total cylinder feed. If the condensation W' can be estimated, either by a formula or by observing the temperature distribution, the remainder $W^\circ - W'$, which may conveniently be represented by the symbol W'', may be most probably attributed to leakage. According to the Authors' experiments, the main part of the leakage at any point of the stroke may be represented by the formula,

$$W'' = L\,(p' - p''),$$

where $(p' - p'')$ represents the difference of pressure between the valve-chest and the exhaust. If the factor L cannot be measured, it may be estimated by the method of p. 33, from the mean overlap and the perimeter of the ports. Since the area of the steam-ports is generally designed to vary as $d^2\,N$, where N is the piston speed, L may be taken as being proportional to $d\,\sqrt{N}$ in similar engines.

Illustrations of Partial Condensation.—As a test of the validity of the method of reducing to the equivalent clearance surface in the case of partial condensation, and as showing the relative unimportance of the barrel surface, the following cases are cited, having been selected by Cotterill (*loc. cit.*, p. 334) for a similar

TABLE VIII.—CONDENSING TRIALS.

Authorities and Trial Mark.	Stroke, *l*.	Diameter, *d*.	Revolutions per Minute.	Cut-off, *c*.	Absolute Pressure at Cut-off.	Missing Quantity.		Equivalent Clearance Surface, S.[1]	Condensation Factor, C.
						Lbs. per Hour, W°.	Per Cent. of Feed.		
	Feet.	Feet.						Sq. Ft.	°
Mair . . (L)	5·5	2·67	20·3	0·26	46	770	29	33	23
„ . . (M)	5·5	2·67	20·3	0·26	46	1,200	37	52	23
Dallas . (D)	2·5	3·0	56·9	0·20	47	1,460	29	60	24
Hirn and (2)	6·5	2·0	30·5	0·26	54	850	30	32	27
Hallauer (5)	6·5	2·0	30·0	0·16	55	450	25	Superheat, 81° F.	
(HH.). . (6)	6·5	2·0	30·4	0·16	55	760	36	29	26

[1] *Calculated* by formula, $S = 2\,(\pi d^2 + l d c)$.

purpose. There is considerable range of speed and of relation of stroke to diameter, but the conditions of pressure are fairly comparable, and the leakage similar, and probably small as compared with the condensation. The engines are also sufficiently large to make the effect of conduction practically negligible.

In the trial Mair (L),[1] the sides and base of the cylinder were jacketed, which would have the effect of reducing the unjacketed clearance surface by about one-third. The condensation constants given by Cotterill for these three cases are, (M) 5·3, (D) 7·0, (HH) 3·4. This wide range of values is probably to be explained by his taking the condensation as proportional to the barrel surface. The agreement of the values of C in Table IX is as close as can be expected, and may be taken as showing that, in the case of partial condensation, the equivalent clearance surface is the better basis of comparison.

TABLE IX.—NON-CONDENSING TRIALS.

Authorities and Trial Mark.	Stroke, l.	Diameter, d.	Revolutions per Minute.	Cut-off, c.	Absolute Pressure at Cut-off.	Compression.	Missing Quantity.	Percentage of Feed.	Condensation Factors.		
							Lbs. per Hour W°.		Equivalent Clearance Surface S.	Actual Clearance Surface.	Factor, C.
	Feet.	Feet.							Sq. Ft.	SqFt.	
C and N (4–6) .	1·0	0·88	74	0·20	95	0·25	43	24	2·6	3·7	17°
Willans . . .	0·5	1·17	110	0·44	44	36°	87	35	4·6	2·8	19°
Simple $\frac{50}{2\cdot 2}$ . .	0·5	1·17	201	0·44	44	33°	93	24	4·6	2·8	20°
C.E. 1893, p. 174 .	0·5	1·17	408	0·44	43	25°	137	19	4·6	2·8	30°
Willans . . .	0·5	0·83	122	0·60	83	17°	55	20	2·4	1·8	23°
Compound $\frac{90}{3\cdot 2}$. .	0·5	0·83	211	0·60	82	13°	54	13	2·4	1·8	23°
Loc. cit. . . .	0·5	0·83	401	0·60	80	—5°	38	5	2·4	1·8	16°
Gately and . (16)	3·5	1·5	68	0·41	65	?	390	11	18·5	?	21°
Kletzsch . (17)	3·5	1·5	69	0·42	50		800	24	18·5		43°
(Thurston) . (18)	3·5	1·5	69	0·40	40		372	16	18·5		20°
Col. English . . Mech. Eng., 1887 . Pl. 90, Fig. 5 . .	1·5	1·33	40	0·30	84	0·00	470	45	12·2	?	39°
J. W. Hill, R.C.	4·0	1·5	75	0·16	99	0·08	1,040	29	16·1	?	65°
(Peabody, H C.	4·0	1·5	76	0·14	100	0·12	1,100	34	16·0		69°
p. 265). Whk.	4·0	1·5	76	0·17	91	0·08	1,124	32	16·2		70°

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[1] *Minutes* of Proceedings Inst. C.E., 1884, vol. lxxix. p. 340.

Table IX illustrates the fact that the results obtained by the thermoelectric method, even when reduced to the equivalent clearance surface, are smaller than any similar results, except perhaps a few of Willans'. The effect of an early compression is also well marked. In the last three cases the steam is recorded as having been occasionally 5 per cent. wet, and the condensation was probably limiting.

Estimation of Leakage.—In the case of partial condensation, it is evident that the result may be so profoundly affected by differences of wetness or of compression, that it would not be justifiable to take the small result found in the case of the Authors' engine, and to assume that the larger results shown by other engines were to be entirely attributed to leakage. With the Robb engine, running double-acting under the same conditions, making allowance for piston convection, etc., the condensation factor ought not to exceed 13 lbs. or 14 lbs. per square foot of equivalent clearance surface per hour. This result is much smaller than that shown by any of the double-acting cases cited. Although this value of the condensation factor cannot be directly applied to the estimation of leakage in other engines, the formula may be of use in comparing different trials of the same engine, or similar trials of different engines, under strictly comparable conditions. In the case of limiting condensation, if the clearance surface and the steam cycle are known, a definite limit to the condensation is at once afforded. Any excess may be reasonably attributed to leakage. Unless the steam is known to be wet, the temperature of the walls should be observed, because it cannot otherwise be certain that the condensation is really limiting. If it happens to be partial the leakage will be under-estimated. If $C = 20$ is taken as being a probable average value of the condensation factor in the case of simple engines, the total loss would be given by a formula of the type

$$W^{\circ} = 40\,\pi\,d^2 + \frac{d\,\sqrt{N}\,(p' - p'')}{10}.$$

which would make the leakage relatively more important in small engines and at high piston speeds and pressures.

Prof. C. A. Smith, using a formula of the type $W^{\circ} = C'd\,(t' - t'')$ to represent the total losses in simple unjacketed engines, measured in lbs. per hour per degree of steam range and per foot of piston diameter, finds values of C' which vary between 1·85 and 4·72. The same series of trials may be represented on the Authors' formula by values of C from 15 to 25, and values of the leakage constant from 5 to 10. This does not appear to be an unlikely range of variation, but unless the condensation is often limiting in simple engines, the leakage must frequently be the greater.

The data required for the exact application of the proposed method of estimating leakage are not generally available in any extant trials; but to exemplify the general conclusions to be derived from the analysis, the triple-expansion trials of the experimental engines at Owens College, described by Osborne Reynolds,[1] may be selected, not only on account of the unimpeachable accuracy of the observations, but also because these engines would appear to have achieved some of the best performances on record for engines of so small a size. If it can be shown that, even in these engines, in spite of their record performance, a large part of the missing quantity is probably to be credited to leakage, it follows, *a fortiori*, that a similar cause may be suspected in more ordinary cases. These trials possess the further advantage of an unusually complete and unreserved record of all the leakage tests which were applied, and of the various minor leakages which were discovered and rectified from time to time.

The trials 41, 35, and 40, in which the receivers were jacketed but not the cylinders, will be taken as an illustration. The effect was probably to dry the steam completely at the low speed for the low-pressure cylinders, but less completely at the higher speeds. The following are the data for feed and receiver-jacket condensation in the three trials:—

CYLINDER JACKETS EMPTY. RECEIVER JACKETS AT BOILER PRESSURE.

Trial number	41	35	40
Boiler pressure, absolute	204	205	201
Feed per hour (hot well)	464	688	1,055
Jacket condensation, lbs. per hour	99	117	158

The following are the data for the three separate engines:—

Engine Number	I.			II.			III.		
Trial number	41	35	40	41	35	40	41	35	40
Expansions, r	2·7	2·3	2·0	2·4	2·4	2·2	2·7	3·0	2·6
Revs. per minute, n	146	229	322	127	215	320	109	184	276
Pressure range, $p' - p''$	137	132	123	45	50	52	20	21	23
Temperature range, $t' - t''$ degrees	83	78	72	68	70	69	123	115	107
Missing quantity, z° per cent.	40	29	22	41	38	30	51	48	32
Missing quantity, w° lbs. per hour	185	200	232	190	262	317	237	332	337
Condensation limit	43	41	39	83	88	88	310	300	285

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[1] *Minutes of Proceedings* Inst. C.E., vol. xcix. p. 152.

The condensation limit given in the last line, in default of the cards, and of the actual area of the clearance surface, is estimated by the method explained. No allowance is made for the variation in the ratio of expansion, but since the three engines were similar, the values estimated may probably be taken as showing the relative order of magnitude of the condensation-limit to be expected in the three cylinders. Taking engine No. III, it would appear probable that at the lowest speed the steam was practically dry, and the condensation consequently partial. At the higher speeds, the drying in the receivers was probably insufficient, and the condensation had nearly, if not quite, reached its limit. In the case of engine No. I, the condensation-limit is obviously insufficient to explain the missing quantity. It is likely that both the condensation and the leakage increased slightly with increase of speed, but the leak appears to have been much more important than the condensation. The rate of leak required would have been between 1·0 lb. and 1·5 lb. per pound pressure per hour. A leak, roughly estimated at 30 lbs. per hour, was observed on one occasion with this valve standing. The rate of leak required for the valve when running would appear not unlikely, considering the type of the valve, and that it was specially designed for a high speed. Part of the leak may have been due to the piston and to the cut-off valve, but the pistons appear from other evidence to have been fairly tight. The valve was probably mainly responsible. In the case of engine No. II, the same observations are applicable. The rate of leak required would have been between 3 lbs. and 4 lbs. This would appear somewhat excessive, but it is recorded that leakage of the cut-off valve was inferred from the cards in this engine, and was rectified at a subsequent date. It is evident that a leak of the kind described as having been discovered in this valve might be expected to increase considerably with the speed, as the amount of leak would depend on the inertia. The rate of leak in engine No. III may have been upwards of 2 lbs., but is evidently much less important than the condensation, owing to the smaller pressure difference on the valve. If the leakage therefore is rejected, the rate of condensation of steam must increase rapidly with the density. Setting aside the experiments described in the present Paper, this would appear a perfectly tenable hypothesis. Unless, however, the surface of the cylinder has been underestimated, the metal in engine No. I seems hardly capable of accounting for the whole missing quantity at the lower speed, even if the temperature range of the surface of the metal were the same as that of the steam; at least, on any reasonable theory of condensation. The inferences drawn from the above case may be taken as typical of a great number of other cases which might be given.

Direct experiments by Colonel English[1] on initial condensation with a portable engine appeared to show a rate of condensation varying roughly as the density of the steam. The method employed was similar to that used by the Authors for measuring the exhaust leak, except that steam was admitted at each revolution to the clearance space at the back end of the cylinder, and that the sides of the cylinder were jacketed. The speed and pressure were considerably varied, but it was assumed in reducing the observations that the condensation per hour varied as the square root of the speed. The effect observed when measured per hour appears to be much more nearly independent of the speed. It also appears to be nearly a linear function of the pressure difference. At a mean density of 0·15 the results observed by Colonel English would correspond to the condensation of about 40 lbs. per square foot per hour, which is nearly four times the value found by the Authors at a higher density in an unjacketed cylinder. These results may be more naturally explained by supposing a comparatively moderate condensation varying as the temperature difference, combined with a much larger leakage varying as the pressure difference. The rate of leak thus required would be about 1·5 lb. per pound pressure per hour in the non-condensing experiments, and a slightly smaller rate in the condensing experiments. It may be observed that this rate of leak is the same as that found in one of the Authors' valves, which was proved to be absolutely steam-tight when stationary. It does not imply that the valve was in bad condition, or that the engine was at fault in any way.

Expression in Terms of the Steam and Feed.—It is frequently convenient to express the condensation and leakage in terms of the indicated feed and also of the total feed. The percentages of indicated feed being designated by y, and the percentages of total feed by z, the following notation is afforded:—

$$y^\circ = \frac{100\ \mathrm{W}^\circ}{\mathrm{W}},\ y' = \frac{100\ \mathrm{W}'}{\mathrm{W}},\ y'' = \frac{100\ \mathrm{W}''}{\mathrm{W}},\ y^\circ = y' + y''.$$

$$z^\circ = \frac{100\ \mathrm{W}^\circ}{(\mathrm{W}^\circ + \mathrm{W})},\ z' = \frac{100\ \mathrm{W}'}{(\mathrm{W}^\circ + \mathrm{W})},\ z'' = \frac{100\ \mathrm{W}''}{(\mathrm{W}^\circ + \mathrm{W})},\ z^\circ = z' + z''.$$

$$z^\circ = \frac{y^\circ}{(100 + y^\circ)},\ z' = \frac{y'}{(100 + y^\circ)},\ z'' = \frac{y''}{(100 + y^\circ)},\ y^\circ = \frac{z^\circ}{(100 - z^\circ)}.$$

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[1] Proceedings of the Institution of Mechanical Engineers, 1887, p. 503.

If V is the piston displacement per revolution, n the revolutions per minute, w the density of the steam at cut-off in lbs. per cubic foot, and r the ratio of the volume of the feed steam at cut-off to the piston displacement, the formula for the indicated feed in lbs. per hour is $W = \frac{60\,n\,V\,w}{r}$. Also $W' = C\,S = \frac{y\,W}{100}$, $W'' = L\,(p' - p'')$. So that, $y' = \frac{100\,C\,S\,r}{60\,n\,V\,w}$, $y'' = \frac{100\,L\,(p' - p'')}{60\,n\,V\,w}$. The piston displacement V is to be taken as $\frac{\pi\,d^2\,l}{4}$ single-acting, and as $\frac{\pi\,d^2\,l}{2}$ double-acting. The value of the ratio $\frac{S}{V}$ is therefore $\frac{4}{l}$ either single-acting or double-acting.

If the rate of leakage L is taken as proportional to the product of the diameter of the cylinder and the square root of the normal piston speed N, $L = L'\,d\,\sqrt{N}$, where L′ may be called the leakage constant, and has values which probably vary between 0·20 and 0·05 in engines of different types.

The formula for the leak percentage y'' may then be written in the simpler form, $y'' = \frac{L''\,r}{d\,\sqrt{N}}$, where L″ is a new constant, which is practically proportional to L′ in all cases which occur in practice of engines running at their normal piston speed. This shows that the effect of leakage on the performance of different engines may be expected to diminish in proportion as the diameter and the square root of the piston speed are increased. If, however, an engine running at its normal speed is being compared with the same engine running below its normal speed, $y'' = \frac{L''\,r\,\sqrt{N}}{2\,d\,n\,l}$, which shows that the effect of leakage on the performance will increase in direct proportion as the speed is diminished.

The formula for the condensation percentage y' may similarly be written, $y' = \frac{C'\,r}{N\,w}$, where $C' = \frac{40\,C}{3}$, and N is the piston speed, normal or otherwise. Comparing this formula with the leakage formula, the effect of condensation on the performance is not diminished by increase of diameter, but is much more affected by increase of piston speed; it is also considerably diminished by increase of initial pressure, which is not the case with leakage.

The formula for the total percentage loss due to both causes is—

$$y^\circ = y' + y'' = \frac{C'\,r}{N\,w} + \frac{L''\,r}{d\,\sqrt{N}}.$$

It is interesting to compare this formula with that of Cotterill, $y^\circ = \frac{C'' \log_e r}{d\sqrt{n}},$ which is intended to represent the results of experiment on the assumption of negligible leakage.

As an illustration of the nature of the consequences involved in the assumption that the rate of condensation of steam varies as the density, the triple expansion trials already cited may be taken. From Cotterill's formula the values of the condensation constant C'' for the three engines at the lowest speeds, are:—No. I, 3·4; No. II, 5·0; No. III, 11·0. As the steam was probably fairly dried by the receivers at this speed, it is difficult to see why the value of the condensation constant should be so very different for three engines of a similar type and speed, unless the rate of condensation of steam does not vary as the initial density.

Summary of Conclusions.—The observed wall-temperature cycles show that the range of surface temperature, and the interchange of heat between the walls and the steam, is determined chiefly by the temperature of the walls in each case, and by the finite rate of condensation of steam. For this finite rate of condensation, the value 0·74 B.T.U. per square foot per second per 1° F. difference of temperature, at 300° F., is obtained, a result approximately equivalent to 2·7 lbs. of steam condensed per square foot per degree per hour.

The form of the wall-cycles, and other evidence show that the law of re-evaporation is the same as that of condensation, and that both are probably independent of the pressure. The amount of condensation in any cylinder can, therefore, be deduced by the observation of the distribution of wall-temperature while the engine is running.

From the form of the law of condensation there appears to be for any cycle a limit of condensation when the temperature of the walls is the time average of the steam temperature-cycle. Under this condition of "limiting" condensation, the re-evaporation is incomplete, and the temperature of the cylinder is maintained by the mechanical rejection of condensed water, so that it cannot fall much below this point.

The condensation observed was far below this limiting value, and the initial steam was probably uniformly dry. But a marked effect due to the condensation of wet steam was observed, which leads to the inference that, owing to the abstraction of heat by the subsequent re-evaporation of the deposited wetness, the condensation must always be limiting in cases where the initial wetness of the steam is considerable.

The case of "partial," as opposed to limiting condensation, is probably more common in simple engines. The amount of initial condensation per hour in this case appears to be nearly independent of the speed and of the ratio of expansion, and to vary little with the initial and exhaust temperatures. Simple approximate expressions are given, deduced from the law of condensation, for the effect of initial wetness or of superheating of the steam, which are probably, together with the degree of compression, the most important factors in determining the result.

Illustrations have been given of the method by which, if the requisite data are available, the condensation at any point of the cycle may be correctly computed by means of the condensation areas on the temperature-cycle diagram. This method requires a knowledge of the extent of the clearance surface, and also, in the case of partial condensation, of the mean temperature of the surface, in addition to the steam cycle. The application of this method may be expected to throw light on other causes of loss, and particularly on the amount of leakage under the actual conditions of running, which, from these experiments, appear to be a much more important source of loss than is generally admitted.

The thanks of the Authors are due to Mr. J. J. Guest, Assistant Professor of Mechanical Engineering, and to Mr. A. W. Duff, Demonstrator of Mechanical Engineering, for assistance in preparing the figures for this paper, and in taking the observations for the measurement of the conductivity of cast iron. Valuable assistance has also been rendered by Mr. H. M. Tory, M.A., and Mr. H. T. Barnes, M.A. Sc., Demonstrators of Physics, and by Messrs. MacDougall, Rutherford, and Laurie, students of Applied Science.

The Paper is accompanied by twenty-four drawings, from which Plate 6 and the *Figs.* in the text have been prepared.

[APPENDIX.

APPENDIX.

VERIFICATION OF THE TEMPERATURE CRITERION FOR LIMITING CONDENSATION.

As a further confirmation of the law of condensation and re-evaporation proposed in the present Paper, it was evidently desirable to observe the temperature of the cylinder-walls of an engine, in some case in which water was undoubtedly present in the cylinder. The temperature of the walls in this case should be the same as the time-mean of the steam temperature. A case of this kind presented itself recently in the McGill workshop engine, which had to be run throttled, when the cut-off valve had been removed for repairs. The whole of the indicator-diagrams, on reduction, showed the temperature of the mean of the steam cycle to have been the same as that of the clearance surface, within the limits of error of the observations, although on some occasions diagrams were purposely taken when the load was suddenly changed. On November 13th the cut-off valve was replaced, and the condensation after this date apparently ceased to be limiting. The following Table gives a number of the results observed. It will be seen that the relation in question appears to hold over a considerable range of temperature so long as the condensation is limiting.

WALL TEMPERATURES IN LIMITING CONDENSATION.

Date, 1896.	Cut-off.	Remarks.	Temperatures of Steam Cycle.				—	Temperatures of Metal.
			Max.	Min.	Range.	Mean.		
			° F.	° F.	° F.	° F.		° F.
Oct. 17	$\frac{3}{4}$	Throttled .	272	228	44	253	Non-condensing	254
„ 17	$\frac{3}{4}$	Half open .	325	268	57	298	„	299
Nov. 24	$\frac{3}{4}$	Throttled .	283	241	42	260	„	260
„ 5	$\frac{3}{4}$	„ .	292	243	49	266	„	267
„ 5	$\frac{3}{4}$	Rising . .	277	232	45	253	„	250
Nov. 6	$\frac{3}{4}$	Falling . .	253	204	49	230	Condensing .	233
„ 6	$\frac{3}{4}$	Throttled .	236	200	36	221	„ .	221
„ 9	$\frac{3}{4}$	„ .	255	212	43	235	„ .	236
„ 10	$\frac{3}{4}$	Rising . .	277	230	47	251	„ .	248
Nov. 13	$\frac{1}{3}$	Half open .	326	253	73	284	Condensing .	297
„ 13	$\frac{1}{3}$	Quarter open	312	232	80	265	„ .	283

Discussion.

Mr. Hawkshaw.

Mr. J. Clarke Hawkshaw, Chairman, in proposing a vote of thanks to the Authors, said the subject dealt with in the Paper was a difficult one, but that there were members present who would be able to discuss it and give the Institution further valuable information on the points referred to.

Mr. Longridge

Mr. Michael Longridge regretted he had been unable, in the time which had elapsed since he had received the Paper, to obtain a clear idea of the Authors' arguments or to form any decided opinion upon their conclusions. The method of making temperature measurements within the metal by thermo-electric couples appeared to have been very ingeniously applied, and he could understand the Authors being so much in love with it as to be led to build, upon the results obtained by it, a superstructure of conclusions perhaps somewhat larger and more elaborate than the solidity of the foundation warranted. Certain discrepancies in these results were, for instance, attributed to leakage past valves and pistons, and this source of error was stated to be much more serious and important than it had hitherto been considered by experimenters. He thought it required stronger evidence than the Authors had adduced to establish their conclusion. On the other hand, they had satisfactorily proved, what had, doubtless, been long suspected, that the temperature of the inner surface of the cylinder did not follow that of the steam, but moved through a smaller range. Also, the temperature curves obtained during compression seemed to settle the controversy between Hirn and Zeuner regarding the presence of water in the clearance spaces of a steam-engine. The point on which information was at present specially required was the relation between speed of revolution on periodic turn and initial condensation, and on this point he had failed so far to extract any information from the Paper. The Authors appeared to hold, p. 4, that cyclical condensation—which meant, he supposed, condensation per revolution—was independent of speed, or, in other words, that the amount of condensation per hour varied as the number of revolutions, while, on p. 52, W′, the condensation in lbs. per hour was said to be approximately independent of the speed, which was equivalent to saying that the condensation per revolution, or cyclical

Mr. Longridge. condensation, varied inversely as the number of revolutions. He hoped this apparent discrepancy would be explained.

Mr. Stromeyer. Mr. C. E. STROMEYER had listened to this Paper with great interest and thought it gave very valuable information on the subject with which it dealt. In his experiments, Mr. Bryan Donkin had used mercury thermometers, which naturally only gave average results. By substituting electric thermometers the Authors had made a decided advance, for they had thus been enabled to obtain a fair indication of the extent of the rapid changes of temperature during each revolution. He should, however, have been glad to have heard more about the reliability of the thermo-electric method, and how it was safeguarded against leakages and other disturbing influences. He had frequently been impressed by the extraordinary sensitiveness attainable with modern electrical instruments; but because they afforded the means of detecting trifling changes of conditions, they required to be most carefully protected from them. Thus, as far as could be seen, the cycle shown in *Fig. 6* differed materially from that shown in *Fig. 7*. The one was obtained when the iron wire was pressed against the cast iron, the other was obtained with the iron wire soldered to the cast iron. In each case the electromotive force ought to be the same, but in the one junction there would be a slight flow of heat, in the other there would be none. Other disturbing causes would readily suggest themselves, and had, doubtless, been guarded against, but as the subject was new, and certainly very important, it would have been instructive to have heard more about them. The Authors' experiments on the conductivity of cast iron had been carried out in the orthodox manner, and he thought that they, relying on the care which they had taken, had unjustly condemned previous experiments. Their metal seemed not quite the same as that used by others. In fact there might even be a difference in the conductivity of the cylinder cover and the 4-inch sample which was experimented upon, for the one would coal very much faster than the other. He would suggest that the cover itself should be used for the necessary determinations, either by utilizing the thermal junctions which were already fitted or by making the following experiment.

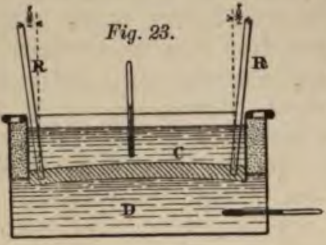
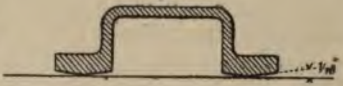


Fig. 23.

Mr. Stromeyer

The rim of the cover would have to be cut off and the remaining flat plate C, *Fig. 23*, surrounded by a thin sheet of metal as shown. The dish thus formed would be filled with warm water and placed over another dish, D, containing cold water. The difference of temperature might be maintained constant for a long time, and the transmission of heat could easily be estimated. As the one side of the plate would be warmer than the other, it would curve as shown, and this curvature, which could be measured by the movement of the rods RR, or by angular displacement of two mirrors attached thereto would give the temperature gradient. This experiment, applied to a different purpose, had been made by Mr. A. F. Yarrow,[1] and Mr. Stromeyer had shown[2] that, with the observed movements of the rods RR, which amounted to $\frac{3}{4}$ inch, the amount of heat transmitted must have been 142 evaporative units per square foot per hour. On the subject of the lagging of the indicators, to which the Authors had referred, an exhaustive Paper[3] had been read at the Institution some years ago, and he had made experiments on it in marine-engine trials,[4] and had found that, with a 6-foot length of cord, the indicator always lagged $\frac{1}{8}$ inch. In other words the upper or steam line on an indicator-card taken under these conditions ought to be moved $\frac{1}{8}$ inch from the end of the stroke. He thought if that were taken into account the amount of condensation mentioned would be materially reduced. The leakage of steam past the slide-valves was perhaps one of the most important subjects mentioned by the Authors, particularly as it was now becoming a general practice to use no lubricants in marine-engine cylinders. He understood that many steamers crossed the Atlantic, or went to other parts of the world, using no oil whatever in the cylinders. A few drops might be applied on starting, and a few drops before reaching port, to prevent the valve sticking fast; and he believed the engines worked perfectly. Whether the leakage under such conditions would be greater or less he did not know; but the practice of doing without oil appeared to work very satisfactorily,

Fig. 24.



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[1] Transactions of the Institution of Naval Architects, 1891, vol. xxxii. p. 108.

[2] "Marine Boiler Management and Construction," p. 94.

[3] Minutes of Proceedings Inst. C.E., vol. lxxxiii. p. 1.

[4] Transactions of the Institution of Naval Architects, vol. xxxi. p. 145, and vol. xxxv. p. 407.

Mr. Stromeyer. at least at sea. In connection with the leakage of steam, he had frequently noticed in very large slide-valves, particularly single-port ones, that their faces did not wear flat nor even round, but were worn in facets. He had noticed this particularly in the case of a single-ported engine, with a very small stroke and consequently very wide ports. In that case the face was out of line fully $\frac{1}{16}$ inch, *Fig. 24.*

Mr. Bryan Donkin. Mr. **Bryan Donkin** had had the advantage of seeing at Montreal the steam-engine described in the Paper, and some of the instruments the Authors had used in their experiments. The Paper appeared to be a masterly treatise on the condensation of steam; and the experiments had evidently been made with some of the best electrical thermometers known. The Authors had described a mass of very delicate experiments lasting one year, and the account now given of them in the Paper appeared the best and most comprehensive yet published on the important question of the temperature of cylinder walls and on the condensation of steam. Great care appeared to have been taken on all points touched upon by the Authors. The electrical method of measuring and determining the temperature of the cylinder-walls and covers as described in the Paper seemed to be much more suitable than the mercurial method, which he had adopted in his experiments. But it was a much more delicate method, and the instruments used—the galvanometers and others—were difficult to observe in a boiler- or steam-engine house. The experiments proved the enormous importance of cylinder-wall temperature compared with that of the working steam—or the difference in temperature between the walls and the steam—in studying condensation and steam-engine economy. They brought out clearly the fact that the clearance surfaces were much more important than the barrel surfaces. It was stated that 90 per cent. of the condensation took place on the clearance surfaces. The method of having a platinum thermometer actually travelling in the steam was novel and very interesting; the temperature of the steam should be very nicely determined by the indications of such a thermometer travelling at a considerable velocity in the moving body of steam. Great importance was attached by the Authors to the slide-valve leakage. With a slide-valve of the type adopted, it was very likely to take place; but he did not think it could be inferred that all valves would leak to that extent. The slide-valve used was very likely to be deflected in the way Mr. Stromeyer had mentioned. But there were lift valves, such as that of the Sulzer

Mr. Bryan Donkin.

type, which probably would not leak. Apart from the question of condensation, the Authors had made an interesting set of experiments on the conductivity of cast iron, which appeared to show the value to be about one-third less than had been hitherto taken—5·4 instead of 7·5. That was a very important result, and he hoped it would be confirmed by other experiments. With regard to the depth to which the heat penetrated in cast-iron walls, it appeared, at 42 revolutions per minute, to be $\frac{6}{10}$ inch. He might mention that, with a slower speed engine, he had found a still greater depth, $\frac{3}{4}$ inch to about 1 inch. The question of time was vital in a matter of that kind. He agreed with the remark of the Authors, that the testing of the leakage of a piston and a valve with the engine blocked and stationary was of little use. It was much preferable to make the engine single-acting, working it with steam on one side and allowing the other side to be more or less open. That had been done by other experimenters, although it was very troublesome. It was acknowledged by the Authors that jacketing, or superheating, or often both, tended to reduce condensation. He hoped that the classical experiments described might be continued, not only with regard to the temperature of cast-iron walls and steam, but with gas- and oil-engines. He had placed upon the table a very simple instrument, called by Hirn the revealer. It was screwed on to the indicator-cock of any engine, and (after it was once heated, which should be very carefully performed to avoid breakage of the glass), the steam passed in and out at every stroke of the engine, and, according as there were one or two glasses, there were different effects of condensation. It was extremely interesting to watch, through the glass, the rapid condensation and the equally rapid evaporation. When the walls were heated only, an extremely fine mist was formed. With different temperatures of the walls the size of the globules increased from fine mist with hot walls to $\frac{1}{10}$ inch or $\frac{1}{8}$ inch with cold walls, which then ran down the surfaces.

Mr. Dolby.

Mr. E. R. Dolby noticed, in regard to the thermal conductivity experiments, that the value the Authors stated as their result, 5·65 thermal units, was for cast iron, whereas "the generally assumed value, 7·5 thermal units, was for wrought iron." As far as he understood, the Authors did not assert that there was any difference between the value they had found and the value previously assumed. He had been much interested in the transmission between the surface of the cast iron and the steam, and

Mr. Dolby. especially so in comparing it with the heat given off from cast-iron hot-water pipes used for heating air. A short time ago he had had occasion to examine the experimental data obtained by Sir William Anderson[1] upon the amount of heat given off from cast-iron surfaces to air. The experiments were made with a difference of temperature ranging from 0° F. to 200° F. It appeared from Fig. 9, Plate 8, in Sir William Anderson's Paper, that, for a difference of temperature of about 200° F., 2 thermal units were given off per degree of difference between air and cast iron per hour, or about 0·033 thermal unit per degree of difference per minute. Comparing that with the statement on p. 20 of the Paper that 6 thermal units of absorption were noted, it showed what a great difference there was between the two. He could not understand why there should be such an immense difference between the amount of heat given off from the surface of cast iron to a gas like air, and the amount absorbed from perfectly dry steam when there was about the same difference in temperature, and he should be glad to hear an explanation.

Mr. Druitt Halpin. Mr. Druitt Halpin had had to deal, 26 years or 27 years ago, with a similar case to that illustrated by Mr. Stromeyer when experimenting with a balanced slide-valve—a valve that was very ingenious, because it could be seen at work. He had taken the valve-chest cover off and had seen the valve working at a pressure of 150 lbs. per square inch. Trouble, however, was caused by the leakage. After successive trials he had at last found a simple method of demonstrating that the valve was out of truth. The valve was scraped dead true to a standard surface-plate, and was afterwards placed with tongs in a bucket of water at 212° F. On placing it back on the plate three or four layers of tracing-paper could be inserted at different places, the valve had so buckled. That was a terrible lesson to show what leakage might amount to. He thought proper caution should be exercised in using the Authors' statement as to conductivity. Fundamental data were given referring to a cast-iron plate 1 inch thick and 1 foot square. The number of thermal units passing through had been derived from experiments made with the apparatus shown in *Fig. 12*, in which he thought the conditions were totally different. If a hole of 1 inch diameter were drilled into a 1-inch iron bottom of a reservoir of water of any depth, there would be a certain efflux of water; and the hole 1 inch in diameter and 1 inch long would be a pipe, and there would be friction in the pipe which had to be overcome by the water. But that friction in the pipe was only an exceedingly small part of the total resistance. There was the resistance of entrance and the resistance of exit, which were exceedingly large in a pipe of those proportions. In the same way the amount of heat that could be received by a plate, transmitted through the thickness of the plate, and then emitted were totally different quantities from the amounts transmitted, through the several sections, 1 inch thick, of the experimental bar shown in *Fig. 12*. Mr. Dru Halpin.

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[1] Minutes of Proceedings Inst. C.E., vol. xlviii. p. 257.

Captain H. RIALL SANKEY wished to add his praise to that already bestowed upon the Paper, especially in regard to the method employed for thermo-electric measurement. He hoped to be able to use it in experiments he was undertaking on the specific heat of superheated steam. The Authors' conclusion that the missing quantity, instead of being largely due to initial condensation, was principally due to valve-leakage, was no doubt true in the engine tested by the Authors. The missing quantity was enormous, and the valves were such that they might be expected to leak very badly; but that all valves would leak to such an extent was a proposition that could not be maintained. About 9 months ago he had commenced experiments on the leakage of piston-valves. They were made with a small Willans engine driven by a motor, so that the valve was in motion as in an engine. The leakage found with piston-valves fitted with the ordinary rings and springs was considerably less than that found by the Authors. On p. 54 the second example was "Willans simple $\frac{50}{2 \cdot 2}$," and the missing quantity was 93 lbs. per hour. He had calculated the leakage in that particular trial from the experiments he had just made, and found that it was between 5 lbs. and 6 lbs.—a very small percentage on the 93 lbs. He also gathered from the Paper that the condensation due to the metal surfaces was comparatively small; and, as he had shown, at any rate in the particular example quoted, that the leakage was very small, it followed that the remainder of the missing quantity was in all probability due to condensation produced by water in the cylinder. He might mention that that was precisely the view Mr. Willans maintained when he read his first Paper—that the condensation in the cylinder was due principally to water and not to the walls themselves. Returning to the valve-leakage experiments, he found that the provisional law mentioned by the Authors (p. 32) $k = \frac{Cp}{l}$ fairly represented his own experiments. Capt. San

Mr. Walker. and factories in England, more was being wasted in steam-pipes and steam-chests than in the engines. If the steam could be as nearly as possible standing still (that was, advancing from the boiler to the work as nearly as possible at the speed which the horse-power to be driven demanded it), and could be drawn off at such a low speed that it did not cause any rush, a great deal more would be saved than by warming the sides of the cylinders or by any ultra-refinements. A previous speaker had referred to leaky valves, and had said that a large part of the loss in an engine was due to the valves bending. He did not think it should be assumed that every valve would bend; he agreed with Capt. Sankey that the design of a valve would settle whether it would bend or not. Considering the immense weights to which engineers were accustomed, it seemed absurd to acknowledge that a valve could not be made of such a shape that, at any rate, the weight at the back of it would not bend it. He believed there was a great deal more to be done in preparing steam at the boiler—that was, sending dry steam out of the boiler, and conveying it along steam-pipes of ample size, so that it was not always moving about and condensing. He thought it was a great deal easier to save money in that way than by extravagant refinements in the engine itself. The refinements would have to be nursed, and it was much better for an engine in the hands of an ordinary workman not to have too many refinements. He could not illustrate his view better than by the ordinary marine Admiralty engine or a great shipping company's engine, in which the very last point on the $\frac{1}{8}$ lb. of water per I.HP. per hour was agreed upon, while the steam-pipes were made of copper. He would rather make the pipes of iron or steel, covering them with a material that would prevent the heat escaping. It was absurd to trouble about the temperature of the wall of the cylinder, and deliberately convey the steam by a pipe which was polished to please the eye instead of being covered to save radiation. At the recent Engineering Conference it had been stated by Mr. Preece[1] that Mr. Benjamin, of Cleveland, had found 50 per cent. or 80 per cent. to be absorbed by shafting in an ordinary factory before the machine to be driven was reached. Without reference to notes, he had, in the discussion which followed, expressed the opinion that 25 per cent. or 40 per cent. would be nearer the mark; and on returning home he had made a trial with an engine which drove 300 yards of shafting.

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[1] Minutes of Proceedings Inst. C.E., vol. cxxx. p. 192.

The shop in which the machinery was placed covered an area of 3,000 square yards. The engine and shafting were running light, driving all the countershafts, and the engine indicated 30 HP.; but 5 minutes afterwards work was started, the engine indicated 120 HP. The loss by the shafting was, therefore, under the most unfavourable circumstances, only 25 per cent. Mr. Wal

Professor W. C. UNWIN said that one lived and learned, and he now heard that one way to improve steam-engines was to enlarge the steam-pipes. He would point out that steam-pipes could not be enlarged without producing two results. The radiating surface was increased and the steam was made to stay much longer in the steam-pipes. He did not think that either of those actions was conducive to conveying dry steam to the engine; at any rate he could quote instances where quite as marked an economy as that which had been mentioned had been achieved not by increasing the size of the steam-pipe but by reducing it. He referred to that matter, however, for a more important purpose. In studying carefully the action of steam in an engine about the middle of the century, Rankine and others came upon this problem; ordinary engines used 50 per cent. to 60 per cent. more steam than there was any trace of on the indicator diagram. The progress of the study of the steam-engine during the last 50 years had been chiefly the study of where that steam disappeared to and why it disappeared. The Paper dealt with that problem—the 50 per cent. or 60 per cent. which went in the working of an ordinary engine not a bad engine but a good one; and he ventured to say that the problem was a more important one than the problem of obtaining local economies by altering the steam-pipes. He considered the Paper was the most important Paper on the steam-engine which had come before any English Society since the first Paper of the late Mr. Willans; and that from the point of view of getting at the bottom of the action inside the steam cylinder, it was the most important Paper ever presented to the Institution. He could quite understand the difficulty which the Paper would present to most readers. One speaker had said that he had not been able to master it in a short time. He was afraid that, so far as he was personally concerned, in 3 months or 4 months he should have to say the same thing, because the Paper bristled so with new suggestions and observations that he was quite sure that 3 months' or 4 months' study would not be too much to enable the whole of the valuable matter contained in it to be understood. It depended entirely for its results on the applica- Prof. Un

Prof. Unwin. An expression of the probable amount of condensation, in a steam-cylinder, working under any conditions, had been found by Professor Cotterill, which fitted a wide range of experiments, with a certain amount of accuracy. He made the condensation depend upon the barrel surface of the cylinder. It was found by the Authors to depend very much more on the clearance surface. That was a question which had already, more or less, arisen, and on which, even after the present Paper, he did not think the final word had been said. Professor Cotterill was so careful in his work than he could not help thinking that he had based his conclusion that the barrel surface was important on a tolerably wide induction. On the other hand, the experiments in the Paper seemed rather conclusive in showing that clearance surface was of more importance. The Authors had certainly given some very valuable results in helping to determine the action of the clearance surface. The Paper bristled with new suggestions. It was, for instance, absolutely new to him to hear of explosive boiling of steam in the cylinder-walls such that water was carried off mechanically without carrying away its proper proportion of heat. Another fact might be of practical importance, namely, that there was evidence of vortical motion of the steam in the cylinder itself, which made the pressure widely different at different parts. There were different pressures in the cylinder at the same moment, so that it was possible that indicator diagrams taken from the side where the pressure was greater than near the centre, might be considerably erroneous. All those suggestions were of remarkable interest, and he thought the Institution was greatly indebted to the Authors for their researches, which must have involved an extraordinary amount of labour and patience.

Prof. Capper. Professor D. S. Capper thought that more information as to the manner in which the practical measurement of the temperatures had been carried out by the Authors would be of interest, particularly as to the calibration of the thermometers. The thermopile appeared to have been connected in the differential experiments with two different points at a small distance apart in a hole sunk in the cylinder; consequently it was the electromotive force produced by the contact between a wrought-iron wire and the cast iron of the cylinder, which had been a measure of the temperature produced at the point at which the contact took place. The only calibration seemed to have been on a bar of metal which was cast at the same time as the cylinder, but as the

Prof. Capper.

Authors had pointed out, the electromotive force produced in pieces of the same metal might differ widely. He would like to know how the temperature had been deduced from the electromotive force produced by the thermopile. Such an explanation would be a great help to members in trying to follow in the footsteps of the Authors. In further reference to the determination of the temperatures, it was stated on p. 10 that the curve in *Fig. 9* corresponded to a range of about 20° F. at the surface of the metal. He was unable to find from the Paper how the temperature of the surface of the metal had been arrived at. If it had been obtained by producing the curve of penetration obtained from the observed results at different depths until it cut the vertical ordinate at the surface, a very wide margin of error might be introduced, owing to the increasing curvature as the curve approached the inside surface. The nearest point to the inside surface that could be reached was about $\frac{1}{100}$ inch from the actual contact of the surface with the steam. It had previously been generally assumed that the rapid alternations of steam temperature in the cylinder were not closely followed by the metal to a greater depth than about $\frac{1}{100}$ inch. He did not, however, mean to suggest that in these experiments a smaller depth ought to have been attempted. Extraordinary difficulties had been overcome with remarkable ingenuity and success, and very great credit was due to the Authors for their truly wonderful results, even up to that point; but he would like to know how that $\frac{1}{100}$ inch had been bridged over in arriving at an estimate of the temperature at the surface. It was highly probable that in reading over the Paper again he would find that the Authors had really satisfactorily explained this point, and that he had misunderstood their meaning, but more detailed information as to calibration of thermopiles and the bridging of this gap would be an immense help. He had always been under the impression that polishing the outside of pipes would prevent radiation, and not conduce to it, as inferred by Mr. Walker. It had also been stated by Mr. Druitt Halpin that conduction in cast iron was totally different when taken along a bar from what it was when taken through a plate; and he arrived at his conclusion by comparing these results with the penetration of water through orifices of different proportions. He failed to realize the distinction, and if he had not known that Mr. Druitt Halpin was sound upon this point he should have imagined that his remarks were based upon an old-time belief in caloric as an absolute substance penetrating through the surface of the metal.

Mr. W. G. Walker.

Mr. W. G. Walker had carried out experiments to compare the effect on the transfer of heat of variously-arranged surfaces projecting from the primary surface of a plate in contact with steam, air and water. Two cylindrical smooth vessels were constructed of exactly similar dimensions, cut from the same brass tube, each $6\frac{3}{4}$ inches long, $2\frac{1}{16}$ inch in external diameter, and $\frac{1}{32}$ inch thick, they were fitted with movable water-tight lids, through which thermometers were inserted. When filled with water, no difference was found to exist between their respective powers to absorb or discharge heat; this was ascertained by heating them together in steam to 212° F., and allowing them to cool either in air or water. The thermometers registered alike throughout the scale. One of the cylindrical vessels was then tried with copper ribs, $5\frac{1}{2}$ inches long, $\frac{3}{4}$ inch wide, and 0·012 inch thick, soldered on longitudinally, the ribs were spaced equally and radially round the cylinder, and were tried internally and externally, as shown in *Figs. 25* (I to IV). The external and internal ribs, when tried together, were in the same plane. The two vessels were heated by suspending them in a tin vessel, *Fig. 26*, 12 inches high, and 8 inches in diameter, having 2 inches of water boiling by a powerful burner; the wooden lid of the tin vessel was movable, and the thermometers passed through it. The two cylinders to be compared were filled with water, and when their temperatures were equal, generally about 65° F., the wooden lid to which they were attached was placed on the vessel containing boiling water; the lid formed a good fit, and the metallic surfaces at once commenced to absorb heat. The reading of the thermometers, together with the time, was noted simultaneously at every 10° of the plain cylinder; the time was taken in seconds by a chronometer, and the temperature to $\frac{1}{5}$°; the thermometers became stationary at 210° F. The lid, together with the cylinders, was then lifted off and suspended either in air or water, and allowed to cool, the respective temperatures and the time being noted simultaneously at every 10°. Contact with steam, air or water always referred to the external surface of the

Figs. 25.

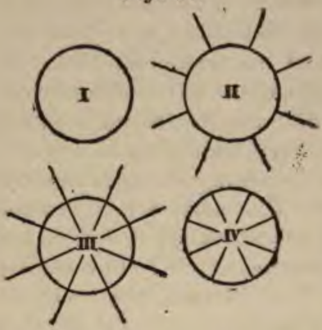
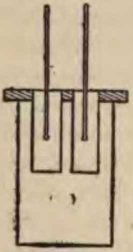


Fig. 26.



cylinders, the internal surface being, of course, exposed to water only. The difference which arose in the conditions between some of the experiments due to the variation in the supply of heat from the burner, change in the atmosphere, temperature and cooling water, did not affect the value of the comparative nature of the experiments, as any change had the same effect on each cylinder. The results of the experiments were given in Tables I–III, p. 80. Mr. W. G. Walker.

The difference between the temperature of the corresponding plain and ribbed cylinders increased from zero and reached a maximum after a certain time, after which they again closed to equal degrees of temperature. The presence of the ribbed surfaces increased to a considerable extent the rapidity of transfer of heat either when absorbing heat from steam or discharging it into air. With the ribbed surface in contact with water the speed of transfer was also increased, but not nearly to so great an extent. In the case of the externally-ribbed cylinder (II), *Figs. 25*, the greatest difference in temperature between the corresponding plain one was 18° F., 33° F., and 8° F. respectively when in contact with steam, air, and water. When exposed for 53 minutes in the atmosphere the temperature of the externally ribbed cylinder had fallen 102° F., whilst the temperature of the plain one had only fallen 69° F. The addition of the internal to the external ribs (III), appeared to produce little or no effect on the result previously obtained. The temperature of the externally and internally-ribbed cylinders showed a maximum difference of 18° F. and 33° F. compared with the plain cylinder when absorbing and discharging heat in steam and air respectively. In the case of the internally-ribbed cylinder (IV), the external ribs having been taken off, the maximum difference in temperature over the plain cylinder was 1° F., 1° F., and 6° F. in steam, air, and water respectively; their outer or external surfaces were, of course, the same in this case. It was interesting to notice the greater effect of internal ribs in the case when the cylinder was cooled in water. The comparatively slow transfer of heat when in contact with steam or air would allow the temperature of the water in the cylinder to adjust itself; when, however, the transfer became rapid, the ribs became more effectual. The external ribs were more effectual in discharging heat to the atmosphere than in absorbing it from steam. This difference might be due to the condensed layer of steam which was deposited on the surface, and which retarded to some extent the transfer of heat. The external surface was also increased by coils of copper wire $\frac{1}{16}$ inch and $\frac{1}{32}$ inch in diameter. With a space between the turns from a

Mr. W. G. Walker.

TABLE I.—PLAIN SURFACE AND SURFACE WITH EXTERNAL RIBS.

Steam.			Air.			Water.		
Plain Surface.	External Ribs.	Time.	Plain Surface.	External Ribs.	Time.	Plain Surface.	External Ribs.	Time.
Temp. °F.	Temp. °F.	Seconds.	Temp. °F.	Temp. °F.	Minutes.	Temp. °F.	Temp. °F.	Seconds.
65	65	0	209	209	0	209	209	0
80	92	35	200	193	3	200	198	10
100	118	167	180	160	16	180	174	30
120	138	295	160	132	30	160	154	65
140	154	430	140	107	53	140	132	110
160	168	590	120	89	84	120	112	170
180	182	757	100	76	132	100	93	254
200	201	950	80	68	255	80	77	480
209	209							

TABLE II.—PLAIN SURFACE AND SURFACE WITH EXTERNAL AND INTERNAL RIBS.

Steam.			Air.		
Plain Surface.	External and Internal Ribs.	Time.	Plain Surface.	External and Internal Ribs.	Time.
Temp. ° F.	Temp. ° F.	Seconds.	Temp. ° F.	Temp. ° F.	Minutes.
65	65	0	209	209	0
80	90	35	200	191	5
100	116	115	180	160	17
120	137	220	160	130	34
140	155	325	140	107	57
160	168	430	120	90	88
180	184	560	100	78	138
200	201	690	80	10	261
209					

TABLE III.—PLAIN SURFACE AND SURFACE WITH INTERNAL RIBS.

Steam.		Air.		Water.		
Plain Surface.	Internal Ribs.	Plain Surface.	Internal Ribs.	Plain Surface.	Internal Ribs.	Time.
Temp. ° F.	Temp. ° F.	Temp. ° F.	Temp. ° F.	Temp. ° F.	Temp. ° F.	Seconds.
65	65	209	209	209	209	0
80	80	200	199·5	200	196	13
100	100	180	179	180	174	43
120	120·7	160	159	160	155	72
140	141	140	139	140	134	118
160	161	120	119	120	115	185
180	181	100	101	100	96	295
200	199	80	79	80	79	640
209	209					

Mr. W. G Walker.

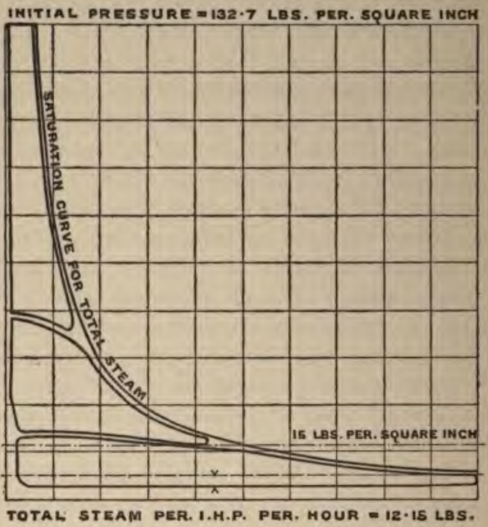
mean of nine experiments, the temperature of the coiled cylinder fell faster than the plain one, reaching a maximum difference of 6·7° F. When, however, absorbing heat from steam the effect was reversed, the temperature of the plain cylinder rising the fastest, and reaching a maximum difference of 4·7° F. This apparent irregularity could be accounted for by the thick layer of steam which was deposited between the wire coils and could not easily drain off. When cooled in water the coiled and plain cylinders gave the same results. In these experiments care was taken that the colour of the respective metallic surfaces was the same. He considered the Paper a most valuable one, especially to himself, because he had had the opportunity of visiting Canada during the past summer, and had seen the apparatus with which the Authors had made their experiments. He was indebted to Mr. Clarke Fisher for having laid upon the table his electrical thermometers, which had been developed at Cambridge by Messrs. Callendar and Griffiths. They depended upon the effect of the source of heat upon the resistance of a platinum wire connected in circuit with a differential galvanometer, from the readings of which the temperature could be directly indicated.

Mr. Dave

Mr. HENRY DAVEY thought the problem dealt with in the Paper was generally acknowledged to be a most complicated one; and, unless the observations had been carried out with extreme accuracy, it would be almost impossible to derive quantitative results which could be used in practice and applied to steam-engines. It was necessary to know the specific heat of cast iron, and experiments were made to determine whether the originally accepted value was true. An error of about 30 per cent. had been found by the Authors, and it had an important bearing on the quantitative results obtained in the investigation. Engineers referred a good deal to the missing quantity of steam arising from condensation in the steam cylinder; but they would more nearly realize the importance of the question if they looked for the missing power. For their standard of reference, the Authors had selected the Clausius adiabatic cycle. Taking the missing power in the best example of triple engine, it would be found that it was not the large quantity that engineers generally imagined. In the best examples, such as that given by Professor Thurston in the accompanying *Fig. 27*, eliminating the error of piston and valve leakage, it would be under 10 per cent. That was the defect to which that law applied. It was a small quantity, therefore any error in the quantity determined by that law would have a very serious effect in its application. With regard to the

Mr. Davey. details of the engine, he could not attempt to criticize the electrical instruments and the thermometry; but, with regard to the engine, he noticed that the valve was not an ordinary slide-valve, nor was it a piston-valve. It was a plate of metal pierced with cavities, assumed to be steam-tight between two surfaces. That was a very old type of valve, and was called an equilibrium slide-valve. It had been used in England 30 years ago, and he believed it was used very largely in America at the present time. In England the experience with valves of that type was the great difficulty of making them steam-tight and keeping them so. Again the results would be vitiated by a difference in the condition of the surfaces exposed to the steam. There were large valve-ports in that engine, and he assumed that the valve-ports were in the condition in which they usually were in such engines, that was, just as they came from the foundry, the sand having been washed off either before the engine was put to work, or after, by the rush of steam, so that it was with what the founders call the skin on the surface. The surface of the piston probably was a machined surface, and probably that of the cover also. He thought definite information was necessary upon those subjects, because the condition of the surfaces affected the rate of condensation very materially.

Fig. 27.



Liversidge. Mr. J. G. Liversidge, R.N., wished to point out one use to which Mr. Donkin's revealer could be applied. There were still many engineers who had a prejudice against the use of steam in the jacket on account of lubrication; they were afraid of having the steam too dry in the cylinders, and consequently scoring them. He had himself made one of the revealers by utilizing a sight-feed lubricator glass, and had fitted it up in connection with the pipes for the indicator, and the action of the jacket was most usefully shown in the revealer. Without steam in the jackets there was far more deposit on the sides of the glass revealer than was the case when steam was in the jacket, but there was still water to be seen on the sides of the revealer even when jackets were in use. He should like to assure Mr. Walker that the Admiralty steam-pipes were very well lagged. If they were not, the temperature in the engine-rooms would be unbearable. He might also mention that steel was displacing the use of copper for large steam-pipes. Mr. Live

Mr. G. M. Clark had had the privilege, some years ago, of assisting Professor Callendar in some thermometric researches which he had carried out in Cambridge. He had, therefore, an intimate acquaintance with the dexterity and skill with which Professor Callendar had carried out that class of research. He thought that some difficulty in grasping the Authors' meaning was due to the compression of their statement, rather than to omissions in the experiments. If the Authors had put their summary of conclusions on the first page instead of on the last, it would have been much easier to follow the general drift of the experiments. An attempt had been made in the Paper to measure directly, by means of thermo-couples, quantities which had been deduced formerly by more or less indirect methods; and, as the value of the Paper depended on how that either carried out or did not carry out previous results, he thought it was of great importance that throughout the Paper great stress should be laid on which were direct measurements and on which were more indirectly calculated quantities. That was specially the case with regard to Table VI, p. 34, where the Authors brought to bear all their results of leakage, and proved that the leakage calculation or leakage correction was a quantity which could be accepted. It was somewhat difficult to see which of those quantities were measured by their new method, and which were old quantities that could be calculated independently. The pith of the whole thing was contained in line 6, but there was nothing to show how that calculation was arrived at. A simple numerical example would have cleared up a great deal of the difficulty. The leakage correction might have been calculated from K = 3, wh Mr. Clark

Mr. Clark. had been given a page or two earlier. Attention had been directed in line 16 to the order of consistency under the different trials, but it would be seen from line 16 that the quantities varied between 0·0074 and 0·0159, but these were not the quantities which caught the eye as being exactly consistent. The greatest was twice the least. He did not know whether that meant that the first three columns were to be compared; if so, he did not think the comparison was of much value, because the experiments were conducted under very similar conditions. The trials were run over a longer time, and were only repetitions of experiments carried out under similar conditions. If another line were inserted treating the means in such a way as to bring the numbers together, it might be much easier to compare them immediately. Any assumption the Authors had made with regard to the leakage coefficient, if they only wished to compare these three columns, was not of very much value. Another example of the want of distinguishing between directly and indirectly measured quantities, though of a rather different nature (which appeared to have caused some trouble), occurred on p. 14, in the measurement of the conductivity and diffusivity of cast iron. The diagram was given, and the result was worked out by the calorimetric method. The Authors had not, however, used that result at all, but had used the second method, and had eventually calculated the conductivity from the diffusivity by multiplying that by the thermal capacity, which had been calculated from the specific heat. No hint was given on that point, and some speakers appeared to have had some difficulty in consequence. One of the principal quantities measured was the ratio $\frac{Q}{A}$. That depended entirely upon the thermal measurement at different depths in the thickness of the walls. There might be some error due to the actual measurement, but he thought that Professor Callendar was too careful an experimenter to fall into such an error. The error arose from the fact that the whole of the thermo-electric circuit was not at one temperature, as part of it must come through the cover or through the walls into the outer air; therefore there must be a thermal gradient and consequent conduction taking place along the thermometer leads, and the temperature of the actual junction would in that way be lowered, to what extent it was impossible to say without more details of the Authors' arrangements than were given in the Paper. Again, if a section of the walls of the cylinder was considered, the temperature gradient across any particular section

Mr. Clark. depended not only on the difference in temperature between the outside and inside, but was also influenced by the gradient across every previous section through conduction along the walls. If the piston was moving from hot to cold, the previous gradients would be all steeper than the particular one chosen. There would, therefore, be, so to speak, a gradient parallel to, as well as perpendicular to, the walls. In extrapolating these from observations taken near the surface to the temperature at the actual surface, perhaps not the real thickness, but some virtual thickness should be taken depending on this compound gradient. That might, to some extent, modify the Authors' conclusions as to the temperature of the surface of the walls, and hence also of the quantity $\frac{Q}{A}$, the rate of condensation. The criticism might be superficial and perhaps not very sound, but it was a significant fact that throughout the Paper the results obtained from observations taken from the cover were much more concordant than from those taken in the side walls. If the Authors' Table III, on page 19, were plotted for $\frac{Q}{A}$ in terms of speed, it would be found that the observations did not lie along with those taken from the cover, and that would be expected from the arrangement of the thermometers. All the cover thermometers were arranged in a circle of 1½ inch radius round the axis of the cylinder, so that they were in such a position that a correction for a virtual thickness would not have to be applied. The central portion of the cylinder cover would scarcely be influenced by the changes in the walls or the changes of temperature in the flanges, whereas those in the side would be more immediately influenced. It was pointed out by the Authors that it was essential that no foreign material should be introduced into the thermo-couple circuit, but on pp. 5 and 9 two places were referred to at which such foreign matter was introduced, first, in the mercury-cup contacts, and secondly, by the tin used in soldering the couples in place. It would be interesting to know whether the formulas given (p. 6) held for both the original and subsequent calibrations of the couples, and to what extent these angles were permanent. The formulas, too, should, for the sake of uniformity, be expressed in the Fahrenheit scale. Again, as by the differential method of working, changes which affected both junctions were not observed, the Authors might more definitely have explained the "small portion of the cast iron of the cylinder," which was used as part of the circuit, especially with reference to freedom of one

Mr. Clark. junction from cyclical changes. In the description of the situation of the thermo-couples, it was not evident where the junction S, 0 inches, subsequently referred to in the Table on p. 19, was placed. Also, it was not clear what the "four similar holes" were "above and below" (p. 10). Were they nearly on a vertical diameter of the cylinder? Again, junction No. 1 side would appear to refer to a couple placed opposite to the clearance space (p. 10) and not to the one previously specified as being situated 2 inches along the cylinder. If, however, "at this point" referred to time and not to place, the meaning of the Authors would be quite different. The word "depth" had been used by the Authors to indicate thickness of metal measured from the inner surface of the cylinder. The mind-picture was that the thermo-couples were inserted in holes drilled in the metal. The word "depth," therefore, bore more strongly on the depth of the hole than on the thickness of metal left undrilled. The Authors had explained their use of the word on p. 6, line 6, which, if understood, might be sanctioned for its brevity. In *Fig. 10* it was shown that the temperature began to rise at each point before steam reached it; why should not horizontal conduction play an important part? On p. 13 it was stated that condensation on the admission surface was due to the lowering of temperature by conduction. It would be of interest to know under what conditions the engine was running whilst the measurements of the outward temperature gradients were taken. If it was at one-fifth cut-off, it would be more natural to put the statement in that paragraph rather than in the next. Again, in the same paragraph, were both the temperature of the "inner surface" and also of the outer surface extrapolated from observations at different depths, or was the outer directly measured? In considering the cyclical variations in the piston, it should be remembered that they were probably of considerable amplitude, as at each stroke the piston-rod entered the atmosphere and the cyclic changes took place through an extended temperature range. In the paragraph, "The Calorimetric Method," it was doubtful whether the Authors wished to convey the meaning that the old value for wrought iron, 7·5 B.T.U., was untrustworthy, or to emphasize that the difference between K for cast iron and wrought iron was also great, as would be expected from the difference in electrical conductivity. The alteration of the brackets from one sentence (line 8) to the next would render matters clearer. In the paragraph headed "Density," it would have been more logical, and at the same time drawn the attention from

Fig. 12, to have said that the values assumed were $c = 4\cdot5$ Mr. Clar
$\frac{Q}{c} = 1\cdot20$, and hence $k = 5\cdot4$. An amplification of the portion dealing with cyclical heat absorption, showing how to advance from actual observations at definite depths to a fully drawn-out temperature-depth curve, to surface temperatures and to heat absorbed, would be of real utility. There were three dotted curves shown on *Fig. 13*; the boundary curves were explained, but the intermediate one was not referred to. The wave-length and heat absorbed might also be expressed algebraically as well as numerically. There was no explanation given as to the separation of Table III into two parts. An example of the method of calculating column 5 should be given, and it might be explained that column 6 was obtained from Table II, column 5, by multiplying the values here given by the ratio column 5, Table III, to 10. In Table VI, line 21 gave not the usual lbs. gross feed per I.HP. hour, but lbs. corrected feed (line 7). This should be noted particularly, as it was an unusual method of stating their consumption in the engine. The usual method would be to use line 5. Column 7, line 21, would then be 56·5 lbs. against 23·8 lbs. It was to be remembered that the engine was designed to run at 250 revolutions per minute, double-acting, whereas in these trials it was run simple-acting and far below the designed speed.

Dr. A. B. W. Kennedy wished to carry Mr. Davey's diagram a Dr. Ken
little further. He would take an extreme case, and suppose that there was no expansion at all, but a perfectly rectangular diagram. Putting 10 per cent. or 20 per cent. or even 100 per cent. more water into the cylinders certainly neither would nor could make the least possible difference in the power, but to have to pay for the additional water (in cost of coal) would, nevertheless, be a very serious matter. He used this extreme case to show that it was not the proportion of the power, but the proportion of the water that was of importance to practical engine users, for here not even the small gain in power mentioned by Mr. Davey would follow from the saving of the whole additional water, but the actual practical gain in cost would follow all the same. The important matter in the present case was the matter of condensation, and it must be measured in condensation, as it had to be paid for, and not in power. He was glad to hear Prof. Unwin's references to Prof. Cotterill. Probably it had not come into the Authors' province to mention him; but his work in connection with the theory of steam-engines in regard to condensation in the cylinder and to the action of the jacket was most notable, as having

Dr. Kennedy. been thought out by a man who, at the time he wrote, had little experimental work to go upon. It was one of the most splendid pieces of work in the theory of the steam-engine, and it was not reduced in importance in any way by the more complete experimental work which the methods of Prof. Callendar and others had rendered possible. He thought it very unwise for gentlemen who had not used electrical methods to throw doubts upon them. Platinum thermometers and electrical resistance instruments generally were at least as easily verified as mercury thermometers. There was some difficulty in verifying either, but the electrical thermometers were correct within a degree of accuracy unattainable by mercurial thermometers. If a doubt was felt as to Prof. Callendar's result, by anyone who was prepared to swear by an indicator card (which the speaker was not), he should refer to *Fig. 18*, showing a curve drawn representing the temperature calculated from the pressures on an indicator diagram throughout the stroke. On that continuous curve were placed a number of crosses, which indicated the temperature measured by a platinum thermometer shown in another *Fig.* It was evident that the readings of the platinum thermometer and the readings of the indicator springs had been practically identical. He himself considered this to be a check upon the indicator springs, not upon the platinum thermometer; but engineers who had a kindly belief in indicator springs, as friends of their youth, might, if they thought proper, regard it as a reasonable corroboration of the platinum-wire thermometer.

Mr. Davey. Mr. H. Davey said he ought to have added "missing power as the result of cylinder condensation." He wished to point out that the missing power, in the best modern engines at least, was a smaller quantity than was usually associated with the large amount of cylinder initial condensation which took place in some engines. In the example he had given he saw no advantage in extending the diagram for the actual engine, as the terminal pressure of the adiabatic would be below the pressure necessary to overcome the frictional resistance of the engine.

Dr. Hopkinson. Dr. John Hopkinson desired to refer to one point connected with the particular methods of thermometry used. On carefully reading the Paper, it appeared that two electrical methods of thermometry had been employed: the thermo-electro method, which had been extensively used by Le Chatelier in France, and the method by resistance, which Professor Callendar had brought to great perfection in England. But on a superficial reading, the second might escape notice. He thought it would have been of great value to engineers if those methods had been described in detail, because he believed that was the first time that either method had been employed for measuring the instantaneous value of rapidly varying temperatures. He believed that these methods of thermometry would be largely employed in future by engineers, because, compared with the mercury thermometer, they were more accurate, inasmuch as they were very much more rapid in their action. In the Authors' experiments it would be impossible to use the mercury thermometer for measuring the temperature at the various points in a complete cycle. That difficulty had been completely overcome by the two electrical methods, and he ventured to think that the value of the Paper would have been considerably increased if a description had been given of methods which, though familiar to many physicists, were unfamiliar to most engineers. Dr. Hopkinson.

Mr. W. H. PATCHELL had encountered the difficulty referred to, but with the assistance of Mr. F. W. Burstall, had been able to simply accomplish with the platinum thermometer experiments which would have been impossible with a mercury thermometer.[1] He thought Mr. Walker would not find the electrical thermometer harder to use, or more brittle than the ordinary glass thermometer. There was now only one missing link. The apparatus had to be used by an experimenter accustomed to electrical instruments, but now, happily, even that difficulty was being overcome. Two indicators had been placed upon the table, and they appeared very much like the steam-gauge, with which Mr. Walker was no doubt familiar. If he would only get a thermometer of the kind described and couple one of the indicators on to it, he would never want another mercury thermometer. Mr. Patchell.

Capt. SANKEY thought it might interest members to know that early in 1892 Prof. Callendar and Mr. Willans had undertaken steam-engine experiments with platinum thermometers, but the lamented death of the latter had prevented the trials from being carried out. Capt. Sankey.

Sir JOHN WOLFE BARRY, President, said the Institution was greatly indebted to the Authors for their Paper, which from all points of view was of great value. It was a matter of regret that they were not present to hear the discussion and to verbally supplement some of their descriptions with the explanations desired by some of the members. Another matter of regret was that neither of the Authors was able to write "Associate," "Associate Mem- Sir J. Wolfe Barry.

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[1] Proceedings of the Institution of Mechanical Engineers, 1896, p. 134.

Sir J. Wolfe Barry. ber," or "Member" after their names. He hoped at some future time their names would be seen with some of those titles added.

The Authors. The Authors, in reply, regretted their inability to be present during the discussion, as it was in many cases difficult to understand the exact meaning of questions in the report. They had been taken to task by several of the speakers for the obscurity and difficulty of the Paper, and for the omission of important details and explanations, for which they were not responsible. The Paper, as originally written, contained many details of physical apparatus and methods, and much explanatory and illustrative matter, which were omitted in publication, owing to the necessity of bringing the communication within the limits of space at disposal. This compression of the Paper had rendered it more difficult to understand, and from a physical point of view, less interesting and convincing. The Authors, however, hoped that the general results would prove to be sufficiently intelligible on careful study, and that they might have the benefit of the doubt in case any obvious and important test appeared at first sight to have been omitted.

The ingenious method of measuring the conductivity of a plate by the curvature, as described by Mr. Stromeyer, could not have been applied to the cylinder cover used by the Authors without destroying it. The two methods adopted in the case of the 4-inch bar were better known, and appeared to be capable of greater accuracy. They had not, however, been content with assuming that the metal of the cover and the 4-inch bar, though cast from the same ladle, possessed identical properties. The temperature cycles had been observed at different depths in the metal between $\frac{1}{100}$ inch and $\frac{1}{8}$ inch. The agreement of these cycles on reduction formed a satisfactory test, not merely of the quality of the metal, but also of the accuracy of the observations and of the validity of the method of reduction.

The indicators used throughout the trials were of the Crosby pattern. They were tested against a platinum thermometer, and in various other ways, with consistent results. The indicating gear was specially fitted and tested for high-speed work. At the low speeds actually employed, it was not possible that the lag should have been of the order suggested by Mr. Stromeyer.

It might be observed on a close inspection of the section of the valve and relief-back shown in Figs. 1, Plate 6, that the probability of bending or buckling of the valve as an explanation of the leakage, was minimised by the symmetrical form of the valve and its

The Authors.

freedom from strain. The surfaces were also tested at intervals on a true plane, and were not found to have undergone any marked deterioration as a consequence of wear or of unequal heating, as suggested by Mr. Druitt Halpin.

The lubrication of the valves of marine engines without the use of oil was readily explained on the Authors' hypothesis with regard to the nature of valve leakage. The Authors had recently been able to verify by direct experiment the conclusion that a large part of the economy obtained by superheating or jacketing, especially in small engines, was due to the great reduction of the possibility of leakage in the form of water. They regarded these experiments as a strong confirmation of their views on the nature of valve leakage, and hoped shortly to be in a position to publish full details of the method employed.

They had not intended to advance the method of testing the piston leakage by running the engine single-acting as original. They were unable, however, to quote any case in which the leakage had actually been measured in this manner, and they were not aware that any previous observers had given a method of deducing the leakage between any two points of the stroke. The reason for making the engine single-acting was that the conditions at the two ends of the cylinder were very different, and that it was impossible to obtain satisfactory observations of the metal cycles at the crank end. The reason for confining attention to non-condensing trials had simply been that the time at their disposal was limited, and they preferred to work out one case as completely as they could.

The experiments of Captain Sankey on the leakage of plug- and piston-valves were regarded by the Authors as an important verification of their formula. It was evident that the leakage of plugs must depend on the fit, and that of piston-rings on the strength of the springs. The dependence of the value of the leakage coefficient C on the width of the fissure was quite clearly stated in the Paper, p. 32. The important points were that the leakage of any particular valve was approximately proportional to the pressure-difference, and independent of the speed, assuming that inertia effects were absent. The Authors had also made tests two years ago on the leakage of the Willans engines belonging to McGill College, which were regularly used for electric lighting. They had followed the method described by Willans of blocking the engine in various positions. This method was certainly unsatisfactory in the case of an unbalanced slide-valve, but might, as Captain Sankey had stated, be trusted to give a fair indication of the state of the piston-rings in a Willans engine.

The Authors. They were grateful to Professor Unwin and Mr. Bryan Donkin both for the generous appreciation they had expressed at the meeting, and for the interest they had taken in the work during their visit to McGill College. That the temperature cycle of the surface of the metal must be different from that of the steam, had been shown by Kirsch and Cotterill on theoretical grounds, to which adequate reference had been made in the original paper. They had, however, assumed too large a value for the conductivity of iron, and they explained the difference in the cycles by supposing that the temperature of the metal surface ceased to fall as soon as re-evaporation was complete. They both assumed that the temperature of the metal surface during condensation practically followed that of the steam. The latter result appeared also to follow from Hall's experiments, and from others which might be quoted, but was in direct opposition to those of the Authors. The explanation given by Kirsch would evidently fail in the case in which water was continually present in the cylinder. This case had been specially discussed in the Paper, p. 36, and afforded one of the strongest confirmations of the views there advanced.

With regard to the desirability of extending these experiments to engines of a larger size, the Authors were glad to be able to state that, with the kind permission of Mr. Wanklyn, the general manager of the Montreal Street Railway, they were engaged in testing a large compound-engine of 3,500 HP., directly coupled to a single generator of 2,500 kilowatt capacity. The engine could be tested condensing or non-condensing, and with or without steam in the jackets on any part of the cylinders or receiver. All the quantities, with the exception of the condensing water, could be directly measured, and arrangements had been made for automatically recording the temperatures of the metal and steam throughout by means of electrical thermometers.

The phrase "cyclical condensation" had been used by previous writers on the subject. It meant simply condensation occurring cyclically, and not condensation per cycle. The temperature cycle of the surface of the metal, as given in Table III, had been deduced by the aid of Table II, assuming simple harmonic formulas. As explained on p. 19, this method might lead to some error in certain cases for the surface range, but the value of the quantity of heat absorbed, upon which the results mainly depended, would be very nearly correct. Kirsch had given several ingenious methods of deducing the heat absorption for a cycle of any form at the surface of the metal. He assumed the surface cycle to be the same as that of the steam. The Authors were unable to make use of his methods, as the form of the surface cycle could not be satisfactorily deduced from the observed cycle at a depth of 0·040 inch in the metal. The Auth

Some not unreasonable doubts had been expressed with regard to the results obtained by means of the electrical thermometers. Dr. Hopkinson and others had asked for further details of the construction and testing. These details had been excised as being of too purely physical a character. The Authors might fairly claim considerable experience in the use of these instruments; they had spared no pains in the testing, and they hoped that the evidence of Dr. Kennedy and others who had spoken in their defence would be taken as sufficient proof of the capabilities of the electrical methods. They hoped, with the kind permission of the Council of the Institution, shortly to publish some of the more important of these details.

The account of the conductivity experiments had been compressed from ten pages into two. This would account for some obscurity in the statement of results. The same apparatus had been used for both methods. Of these, the second method was the most accurate and direct, because it most closely corresponded with the actual propagation of heat-waves taking place in the metal of the cylinder itself. The Authors were inclined to agree with Mr. Clark that the headings of the different Tables were not sufficiently clear. Nearly ten pages of space, however, had been saved by omitting the explanations given in the original. The agreements referred to in Table VI were between the observed and calculated values in lines 16 and 17 respectively. This agreement proved that if the calculated value of the condensation were correct, the leakage must be very nearly independent of the speed. The subsequent remarks of Mr. Clark referred chiefly to matters of taste and expression, which could not be profitably discussed at that stage. He had succeeded in finding the correct answers to most of his own questions, and the Authors hoped that time would resolve the remainder of his difficulties.

With regard to the effect of the state of the surfaces discussed by Mr. Davey, the Authors had explained on p. [illegible] of the Paper, that, according to their views, the amount of condensation was limited chiefly by the possible rate of condensation of the steam, and that the state of the surfaces was, therefore, of less consequence than was generally supposed. In confirmation of this view, they had made experiments with surfaces of various kinds

The Authors. from cast iron to polished platinum, and with various amounts of adherent water. Some of these experiments had been recently communicated to the British Association, and had already been published in *Engineering*.

Correspondence.

Prof. Ewing. Professor J. A. Ewing congratulated the Authors on the immense amount of important and novel matter which the Paper contained. It was a long Paper, and, notwithstanding that, the amount of information it contained was so considerable that being packed into a relatively small compass it became somewhat hard reading. Especially was this the case towards the end, where the general conclusions were stated. It was difficult for the reader to realize the great quantity of experimental work which the Authors had succeeded in accomplishing. Much remained to be done, and in several places the Paper must be held to open questions rather than settle them; but, as it stood, it was undoubtedly the most important contribution which had yet been made towards the elucidation of that difficult matter—the action of the cylinder walls on the steam. It was all the more interesting that in several respects it conflicted with existing notions on the subject. On one matter of primary importance, namely, the time rate of condensation of steam on a metallic surface, he understood that one of the Authors had recently been making experiments by a more direct method, and it would be an advantage to have the results of these added to the Paper. He would be glad of fuller information as to the data from which it was inferred that the rate of evaporation was conditioned in the same way as the rate of condensation. It would be interesting also to know whether the rate of condensation was affected, to any considerable extent, by the presence of air mixed with the steam. It was at first sight somewhat startling to find the steam in the cylinder superheated during the admission part of the stroke; but that was no doubt compatible with the presence of a quantity of water on the exposed surfaces. The condition of the mixture was then far from one of equilibrium, and it was conceivable that the main volume of the freshly-admitted steam should show superheating, while at the same time there was a layer of water condensed on the walls, so that the substance taken as a whole was wet. It would be very interesting to know whether the apparent fall of temperature during expansion, to a value below that which corresponded to the indicated pressure, was due to a local reduction of pressure through [illegible] and [illegible] was [illegible] exhaust and compression [illegible] amount of condensation [illegible]. The [illegible] lead to an [illegible] incoming steam [illegible] would exceed the [illegible] Authors were [illegible] was very [illegible] were made [illegible] that such a leakage was [illegible] engine [illegible] in amount that [illegible] for it all, and [illegible] accounted for by condensation. [illegible] to insist on the [illegible] estimates of condensation [illegible] that as well as in many [illegible] the highest importance. The Paper was [illegible] scientific analysis, and it was [illegible] mentalists would [illegible]

Professor [illegible] remarked [illegible] steam-engine would never be [illegible] dynamics unless supplemented by research [illegible] ducted on the lines of [illegible]'s investigations [illegible] experiments on these lines afforded a valuable [illegible] subject, even if it should be found that [illegible] affected by an unusual leakage of steam at the valve. The Authors [illegible] appear to regard the leakage as normal, but the [illegible] would hardly be accepted by engineers as applicable to steam-engines in general. In the compound engine at University College, Dundee, the leakage of the valves had been repeatedly measured, and it was difficult to conceive that the observed leakage, which was always a negligible quantity when the valve was stationary, would be increased a hundredfold or thousandfold by setting the valve in motion. Moreover, it was obvious that if the Authors' correction for leakage were applied to the recorded results of Mr. Willans and other observers, the "missing quantity" would wholly disappear from the account, and in many cases the steam admitted from the valve-chest would appear as a smaller quantity

Prof. Fidler. than the steam present at cut-off, and the trials would show initial evaporation instead of initial condensation. But in spite of this accidental defect of the engine, the Authors had obtained results which could not fail to be of interest to engineers who had attempted to devise a quantitative method for estimating condensation in the cylinder. The question was beset by innumerable difficulties, and while condensation was in progress at one part of the surface it was probable that evaporation had already begun at another. Confining attention to an unjacketed clearance surface, and neglecting the barrel or replacing it by an equivalent augmentation of clearance surface, a few equations, which must be true in all cases, could at once be written. Thus, after due allowance was made for the loss of heat by radiation, the metal might be brought into a debtor and creditor account; and if a denoted the area contained between the two extreme phases of the Fourier curve, it would appear that the heat received from the steam at one part of the cycle and afterwards returned, must be $h_1 = a\,c\,F$, where F was the area of the surface. Whatever ambiguity might exist as to the exact form of the curve, this consideration was yet sufficient to show that in most cases the extreme range of temperature in the surface-temperature cycle must be far less than in the steam-temperature cycle; for if it were not so, the quantity h_1 would suffice to effect a much greater condensation than was generally observed. If the mean indicator diagram for a prolonged steam trial was analysed at a succession of points along the curves of compression and expansion it would give further indications as to the temperature changes at the wall-surface. By this analysis it was possible to fix, within close limits of error, the beginning and the end of each period in the alternating interchanges of heat —points which must apparently coincide with the two intersections of the curves of surface-temperature and of steam-temperature. And, at several points, it was possible also to calculate the varying rate at which heat was being transferred between the steam and the metal—or at all events between the steam and some external body whether metal or fluid. Plotting as abscissas the successive angular values, θ, of the crank motion, and as ordinates the calculated differentials $\frac{d\,h_1}{d\,\theta}$, a pair of curves might be traced, or partly traced, whose areas must, *ex hypothesi*, represent the quantity of heat h_1 as found by the actual trial. This curve, so far as it could be traced, would represent observed facts which were not dependent upon debatable opinions; and although the curve could not be completed, at least its length, its average height, and a few of its actual ordinates, were known. Then if it was true that the heat-loss (shown by analysis) had really passed into the metal, the following relationships must be exhibited:—(1) The area of the curve above mentioned must represent the same quantity of heat as the area, *a*, of the Fourier curve. (2) The ordinates of the first-named curve must everywhere be proportional to the inclination of a tangent drawn to the corresponding Fourier curve at the wall-surface. Up to this point the equations were independent of any opinion that might be formed as to the causes which govern the rate of transfer between steam and metal; but again, if it was true that the rate of heat-transference was simply proportional to the instantaneous difference of temperature between the steam and the wall-surface, then the following relationships must also be exhibited:—(3) The observed ordinates $\frac{dh}{d\theta}$ should be all proportional to the temperature-differences or to the height intercepted between the curves of surface-temperature and steam-temperature; so that the surface-temperature curve could be (in part) constructed by simple addition or subtraction, or the curve $\frac{dh}{d\theta}$ constructed from the observed cycle of surface temperatures. (4) Drawing at each successive phase of the cycle a tangent to the Fourier curve at the wall-surface, these tangents must all radiate from a single point so long as the pressure of the steam remained constant—as it did approximately during the period of admission, though not very closely. (5) If it were possible to assume, further, that the restoration of heat was governed by the same law, then the two loops of the curve must either have equal areas or must exhibit only such a difference of area as could be attributed to known external gains and losses. With all these simultaneous equations in view, it was possible, without insisting on their accuracy, to see that the fluctuations of surface-temperature must have certain phases and must be confined within certain limits of probable deviation. It might be interesting if some such comparisons could be made between the related quantities in the experiments recorded in the Paper; but it would be hopeless to attempt the analysis of a mean diagram if it were found (as in this case) that nearly nine-tenths of the steam missing at cut-off had to be written off as the probable leakage at the valve. From the imperfect, fragmentary evidences that could be obtained from experimental curves of $\frac{dh_1}{d\theta}$, it did not appear possible that the rate of transfer could depend wholly and simply on the tempera-

Prof. Fidler.

Prof. Fidler. ture-difference. It seemed to depend also on another variant, which might be either the density or the dryness of the steam.

Mr. FitzGerald. Mr. MAURICE F. FITZGERALD regarded the Paper as a valuable contribution to experimental knowledge of the temperature cycles in steam-engine cylinders, containing, as it did, a large number of observations on points as to which, up to the present, limited experimental information was obtainable. The Authors' observations might even be described as the first real attack, except that of Professor Hall, on the problem of directly measuring, with reasonable exactness, the cycle of temperature in a steam-engine, for either metal or steam; mercurial thermometers being hopelessly sluggish for the purpose. The first general conclusion of the Authors, that the rate of condensation on the surface of the metal was simply proportional to the difference in temperature between the steam temperature and the temperature of the wall, seemed to be in fair accordance with the results of Mr. Bryan Donkin's and Colonel English's experiments.[1] The agreement was not close for small differences of temperature, of 3° or 4° only, between metal and steam, but with differences of 10° to 20° it appeared that condensation took place at rates, in Mr. Bryan Donkin's and Colonel English's experiments, not differing notably from that inferred by the Authors. But, on the other hand, it appeared anomalous that there should be this agreement when, as *Fig. 17* would appear to show, the actual contact-difference of temperature between the steam and metal differed extremely from that between the general body of the steam and metal, and the "condensation area," measured in *Fig. 17*, from the dotted curve, would be about three times as large as that obtained from the full line. On comparing the period of heat absorption in *Fig. 14*, with the position occupied by the "condensation area" of the full line in *Fig. 17*, the time-temperature dotted curve of which resembled that of *Fig. 14*, it would appear that the "condensation area" actually occupied, as fairly as might be judged, the part of the revolution in which heat absorption took place; and if the law of proportionality put forward were the true account of the matter, he would expect the results to agree with a condensation area estimated from the dotted curve of *Fig. 17*, rather than the other. There was, therefore, reason to suspect that the law put forward might be really the result of a compensation of errors, and would not hold so nearly true in other engines. But whatever might be the correct ex-

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[1] Proceedings of the Institution of Mechanical Engineers for November, 1896.

Mr. FitzGerald planation, he thought the presence of the layer of steam indicated by *Fig. 17*, must be regarded as a discovery of the Authors, and, unless it proved to have been a mere pocket of steam in the $\frac{3}{8}$-inch hole referred to, a discovery of some importance. He did not think the presence of a layer of steam, with metal on one side and steam on the other, both many degrees colder than itself, towards the end of the lead, could have been anticipated as probable, or, in the absence of distinct evidence of its existence, even credible. He had examined a good many diagrams some years ago,[1] and in all it appeared that, taking the steam on the whole as saturated up to cut-off, a large amount of condensation must have occurred during admission, and no moderate amount of superheating would modify this result, while special experiments made on a large Corliss engine, in which the indicator showed only the part of the diagram belonging to compression on an extended horizontal scale, appeared to show that, reckoned as saturated steam, there was little or no change in the quantity of steam present during compression till the admission-valve opened; the slight apparent variation of the weight of steam present being probably fully accounted for by superheating. There was thus evidence against the highly superheated layer, found by the Authors in their engine, existing after admission began, as it was difficult to conceive how condensation could go on so long as it existed. *Fig. 17* appeared to be a complete overthrow of all the usual expressions as to sudden condensation on cold surfaces during lead and admission, and so forth, commonly regarded as almost axiomatic, and he would expect that it would with difficulty be accepted as a true record of facts. He suggested that condensation might take place under and through a superheated layer of the kind if the vapour near the metal surface consisted, like a gas, of molecules having velocities differing widely among themselves, through having a definite average energy, and that the surface of condensation selected the slow-moving molecules for absorption into itself in a marked manner. A room full of flies could be imagined, some active, others lazy, the walls papered with sticky fly-catcher, which captured the lazy flies near them so quickly that an observer near the wall would find that the flies in the air there were nearly all moving quickly. If the flies were considered as steam molecules, and the fly-catching paper cold metal, the description applied

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[1] Report of the British Association for the Advancement of Science, 1888, p. 819.

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Mr. FitzGerald. to the case. The principal objection seemed to be that no evidence of the existence of a high-temperature layer next a condensing surface, while steady condensation was in progress, had ever been produced; but it did not appear to have been ever looked for deliberately. The remarks on pp. 28–29, on the dynamical conditions of equilibrium of expanding steam, were interesting. The difficulty of dealing with the events which took place during the transition from a state of things in which the kinetic theory of gases applied more or less perfectly, to one in which drops, large enough to have surface-tension, but not so large, perhaps, as to be immeasurably removed from molecular dimensions, was very great, both theoretically and experimentally. It could, however, be shown, without difficulty, that if drops were supposed to exist, of less than microscopic size, somewhat larger than the limiting size of statical equilibrium, the work done against surface-tension in enlarging them by condensation might absorb so much of the heat of condensation set free as to leave little to supply the balance referred to by the Authors, unless the rate of condensation were extremely high, and, with drops small enough, the possible rate of condensation might be sensibly modified by the number of molecules encountered per second being in reality limited. The painstaking care of the Authors, in making the troublesome calculations necessary for verifying the statements as to the precise effect on the surface-range and heat absorption of the peaked curve, *Fig. 14*, as compared with the simple harmonic one, *Fig. 15*, as well as in other points, like the measurement of the conductivity of the cast-iron used, rendered the Paper particularly valuable. The investigations as to valve- and piston-leakage seemed also worthy of further pursuit, as the conclusions arrived at seemed hardly to tally with the circumstance that stuffing-boxes, at any rate, could be made "drop-tight" under hydraulic pressures much heavier than any steam pressures. He had not any clear or decisive experience to prove or disprove whether sliding-surfaces of metal could be made equally tight. Different forms of fitting-surfaces, and kinds of relative motion, greatly affected the tendency to draw in or squeeze out a liquid film between them. This was shown by the great difference in behaviour of cylindrical shaft bearings and end- or collar-bearings, so that it would be unsafe to dogmatize on the matter in the absence of direct experiment, but the general behaviour of slide-bars and blocks, in respect of lubrication, when grooved at sufficiently close intervals, was consistent with the Authors' hypothesis.

Prof. Gu
muth.

Prof. M. F. **Gutermuth**, of Darmstadt, thought the trials described in the Paper not only led to most important results, as to the interchange of heat between the cylinder-wall and steam in the experimental engine, but gave a practical basis for the judgment of the similar proceedings in other steam-engines under different circumstances. He regarded the experiments as a marked advance towards the explanation of the complicated process in the steam-cylinder when at work.

Prof. Jac

Prof. D. S. **Jacobus**, of Hoboken, N.J., considered the approximate formula, by which a condensation factor C had been derived, of special interest. A similar factor had been deduced for tests made by Prof. Denton and himself in 1888 on a simple engine of 17-inch bore and 30-inch stroke, running non-condensing. Any formula for the condensation must necessarily be approximate. In the Tables giving the results of their tests, a factor was deduced which represented the heat-equivalent of the condensation per stroke divided by the product of the time of admission, the surface exposed, the point of cut-off, and the difference of temperature between the steam admitted to the cylinder and the steam at release.[1] This factor had been suggested by Rankine, in discussing the famous experiments of Isherwood on the steamer "Michigan," which afforded the first systematic data for the appreciation of the important influence exerted by the phenomena of cylinder-condensation upon economy.[2] The tests to which he referred were specially adapted for determining the constancy of such a ratio for a given engine, as the speed varied between 9 revolutions and 70 revolutions per minute, and the cut-off from 7 per cent. to 60 per cent. of the stroke; the variation in IHP. being from about 10 to 150. These factors were found to vary considerably and to be largest for a given cut-off when the speed was highest. In studying these figures, it seemed probable, that, if the condensation were assumed to be proportional to the time of admission raised to some power less than unity, the results would fall more in line with each other. This was equivalent to assuming that the rate of condensation at the instant that the steam struck the metal surface was greater than the rate of condensation at any succeeding interval and decreased as some power of the time. The results of all the experiments, except those in which the steam was throttled, and some special experiments in which there

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[1] Transactions of the American Society of Mechanical Engineers, vol. x. p. 722.

[2] Transactions of the Institution of Engineers in Scotland, 1861–62.

Prof. Jacobus. was a very low boiler-pressure, were employed to determine the exponent of the time. This exponent was found to be 0·68 in their tests, and the missing quantity in British thermal units per stroke, divided by the product of the temperature raised to this power, the surface in square feet to cut-off and a range of temperature from admission to the end of expansion, was found to be much more nearly constant than the simple factor obtained by dividing by the time, surface and range. Designating the missing quantity in British thermal units per stroke by Q, the time of admission by t, the exponent of 0·68 by y, the surface to the cut-off by S, and the range of temperature from admission to release by R, the average results of the experiments gave $Q \div t^y S R = 0 \cdot 29$. The highest value of this ratio was 0·41 and the lowest 0·22. A similar ratio was then obtained for Willans' tests, where the low-pressure cylinder was run as a non-condensing simple engine, and the average of all the tests was found to be 0·31; the highest value being 0·4 and the lowest 0·18. This was a remarkable agreement for two engines of such distinct types, the Willans engine being of 14-inch bore and 6 inches stroke, running at 400 revolutions per minute; whereas the engine on which their tests were made was of a much larger size and ran at a low speed. The factors for each of these tests were given in col. 10, Table 1, and the factor C, described in the Paper, in col. 12. An examination of Table I, p. 103, would show that the factor C varied to a greater extent than the factor given in col. 10. The factors for Willans' steam-engine trials were given in Table II, p. 104, which showed that the same remarks applied to the constancy of the two factors, as in the case of their tests, the ratio of $Q \div t^y S R$ being more clearly constant than the factor C. If the pressure at release was greatly different from that of the atmosphere, the missing quantity divided by $t^y S R$ might be considerably different from the value of 0·3 here deduced, which might be assumed to apply only to unjacketed engines cutting off at less than 0·6 stroke and exhausting against atmospheric pressure. The time of admission was taken as the time required for the crank to move from 10°, before the dead-centre point to the point of cut-off; or, if α be the angle from the dead centre to the point of cut-off and N the number of revolutions per minute, the time of admission would be $\frac{\alpha + 10}{6 N}$. The surface was taken as the clearance-surface, and that portion of the surface of the cylinder and piston-rod exposed to the point of cut-off. He noted that the leakage correction, given in Table VI, p. 34, was greater than the corrected feed-water used by the engine. Although the experiments determining this leakage appeared to have been

Prof. Jac

TABLE I.

Simple engine—17-inch bore; $29\frac{3}{4}$-inch stroke; rods $2\frac{9}{16}$ inches and $2\frac{7}{8}$ inches in diameter.

Clearance surface = 5·81 sq. feet. Average clearance volume = 6·3 per cent.

Number of Test.	Cut-off.	Revolutions per Minute.	Missing Quantity.		Time of Admission in Seconds t.	Surface to Cut-off in Square Feet S.	Range of Temperature from Admission to End of Expansion R.	t^y S R $y =$ 0·68.	Missing Quantity in B.T.U. per Stroke ÷ t^y S R.	Equivalent Clearance Surface S^1.	Factor C.
			Lbs. per Hour.	B.T.U. per Stroke.							
1	2	3	4	5	6	7	8	9	10	11	12
1	0·599	70·38	695	72·5	0·264	13·48	40·4	220	0·33	16·85	41
2	0·599	61·55	570	68·0	0·302	13·48	40·4	241	0·28	16·85	34
3	0·599	25·69	466	132·9	0·724	13·48	36·4	394	0·34	16·85	28
4	0·599	17·34	410	173·5	1·073	13·48	34·6	489	0·35	16·85	24
5	0·599	12·63	403	234·1	1·473	13·48	32·8	575	0·41	16·85	24
6	0·599	10·27	291	207·5	1·811	13·48	32·7	660	0·31	16·85	17
7	0·599	8·93	287	235·6	2·083	13·48	33·3	739	0·32	16·85	17
8	0·313	87·60	705	59·3	0·148	9·82	79·7	214	0·28	14·83	48
9	0·313	61·85	672	80·4	0·210	9·82	75·5	257	0·31	14·83	45
10	0·313	58·02	656	82·9	0·224	9·82	75·6	269	0·31	14·83	44
11	0·313	28·55	479	123·3	0·455	9·82	72·6	417	0·30	14·83	32
12	0·313	16·06	367	167·2	0·809	9·82	71·5	608	0·27	14·83	25
13	0·313	13·07	329	184·8	0·993	9·82	66·7	653	0·28	14·83	22
14	0·313	8·63	270	229·6	1·506	9·82	61·3	795	0·29	14·83	18
15	0·182	86·00	680	58·5	0·117	8·14	99·9	189	0·31	13·90	49
16	0·182	64·10	644	74·3	0·158	8·14	99·1	230	0·32	13·90	46
17	0·182	60·88	480	58·3	0·166	8·14	98·9	237	0·25	13·90	35
18	0·182	59·94	573	70·7	0·168	8·14	98·8	239	0·30	13·90	41
19	0·182	59·76	648	79·6	0·169	8·14	97·9	238	0·33	13·90	47
20	0·182	25·62	355	101·2	0·394	8·14	92·6	400	0·25	13·90	26
21	0·182	23·66	370	114·5	0·427	8·14	91·9	420	0·27	13·90	27
22	0·182	17·30	304	128·5	0·584	8·14	89·9	508	0·25	13·90	22
23	0·182	13·30	270	148·8	0·759	8·14	87·5	590	0·25	13·90	19
24	0·182	13·13	293	163·8	0·769	8·14	87·2	593	0·28	13·90	21
25	0·182	13·00	299	169·0	0·777	8·14	88·2	605	0·28	13·90	22
26	0·182	10·50	279	194·5	0·962	8·14	81·7	648	0·30	13·90	20
27	0·182	9·98	249	183·1	1·012	8·14	90·7	744	0·25	13·90	18
28	0·182	8·64	266	225·9	1·169	8·14	84·5	765	0·30	13·90	19
29	0·126	62·80	504	59·4	0·137	7·43	109·0	210	0·28	13·50	37
30	0·126	60·07	486	59·2	0·143	7·43	108·0	213	0·28	13·50	36
37	0·599	62·51	482	57·6	0·298	13·48	36·7	217	0·27	16·85	29
38	0·313	62·66	470	56·8	0·207	9·82	69·7	235	0·24	14·83	32
39	0·182	59·60	392	49·6	0·169	8·14	89·8	219	0·23	13·90	28
40	0·126	60·10	391	48·7	0·143	7·43	102·0	202	0·24	13·50	29
41	0·069	59·00	298	38·0	0·115	6·70	110·7	171	0·22	13·10	23
							Average		0·29	Average	30

Prof. Peabody. that high-speed engines with piston-valves or balanced valves often showed very poor economy in their use of steam. These tests showed very clearly that the condition of the surface exposed to the entering steam had much to do with the effect that surface had in producing condensation. On p. 10 it appeared that the fluctuation of the surface temperature of the head was 4°·9, while that of the adjoining barrel surface was 13°·5. Again, on p. 22 it appeared that nearly equal surfaces at the side of the head and the side of the barrel were accredited with 79 thermal units and 123 thermal units per minute; that was, the metal at the side was nearly one and a half times as active as that of the head. Now the surface of the barrel was polished and kept clean by the rubbing of the piston, while that of the head was probably merely turned, and must have been more or less fouled with grease and dirt. Considering the surfaces of the ports, that were commonly cored out in the casting and were afterwards cleared from sand as well as might be by pickling or otherwise, was it not probable that they were less active than the surface of the head? It appeared probable that all investigators of condensation had over-estimated the influence of the clearance surface, unless these same surfaces were considered to be steam-jacketed; then they were assumed to be entirely innocuous. The conclusion that it was unnecessary to jacket the barrel of a cylinder provided the heads were jacketed did not follow from these tests. Such a conclusion required tests with and without steam in a jacketed engine, and such tests showed just the contrary. For example, three series of tests on the engine in the laboratory of the Massachusetts Institute of Technology,[1] with steam in the jackets on both barrels and heads, with steam in the jackets on the heads only, and with steam in none of the jackets, showed that jacketing the heads only gave a gain of about 3 per cent., while jacketing both barrels and heads gave a gain of about 14 per cent. The Author reported that the indicators were tested as nearly as possible under the conditions of the trials, but did not give the probable errors of the indicators except as incidentally they concluded they were less than the difference between the temperatures of the steam as inferred from the indicator diagrams, and as measured by the platinum thermometer. The maximum error of the thermometer appeared to be not more than 0·5° and the discrepancies referred to appeared to be 2° or 3°, which at 300° F. corresponded to about as many lbs. per square inch—an error

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[1] Transactions of the American Society of Mechanical Engineers, vol. xvi.

Prof. Peabod

which might fairly be attributed to the indicator unless that instrument was tested by some method that was more reliable that those used by the makers.

Mr. Pullen.

Mr. W. W. F. Pullen noticed that the Authors' piston platinum thermometer was first used in a hole in the cylinder cover, and that the dotted curve, *Fig. 17*, represented the variation of temperature it indicated in that position. It would be interesting if the Authors could give data from which a drawing of the thermometer in the position stated, together with that portion of the cover in the immediate neighbourhood of the thermometer, could be constructed, as the indicated temperature of the steam varied so widely from that obtained when the thermometer was placed in the piston. It would appear that the steam in the tube surrounding the thermometer, when fixed in the cover, was highly superheated near the end of compression, while only a few degrees of superheat were noticed when the thermometer was fixed in the piston and occupied a similar position at the end of the stroke. It was not mentioned whether steam was admitted into the jacketed cylinder cover during any of the experiments, and what was the difference in the platinum thermometer curve when steam was and was not admitted to the cover. He believed that a more extended description of the Authors' very beautiful platinum thermometer would be much appreciated.

Prof. Smith.

Professor Robert H. Smith desired to add his tribute of admiration of this Paper and the experimental work it described, which he regarded of the highest scientific character. He had long considered the method of thermo-electric measurement of the variation of cylinder temperature as the only method capable of giving accurate results, but it involved many instrumental obstacles. These difficulties were not even indicated in the Paper. Attempts to use the method had previously been made both at Harvard and at Sibley College, but without substantial success. The greatest merit of the present work lay in the fact that special experimental dexterity seemed to have overcome these obstacles, The electrical registration of a slowly changing temperature had for long been easy; the problem of accurately following and registering a very rapidly varying temperature was wholly different, and immensely more difficult. He hoped, therefore, the Authors would furnish a detailed account of their apparatus and mode of using it. The method of the "condensation area" appeared very successful. Its principle, that of finding the time-average of the difference of temperature between the steam and the metal, and multiplying this by the area of the metal surface

Prof. Smith. exposed to this difference, was the same as that used in the final calculation he had given[1] in 1888; but at that time neither the average metal-surface temperature nor its variations had been measured. The "limited rate of condensation" appeared to be due largely, if not wholly, to surface resistance to conduction of heat. In Peclet's and all other rules for the conduction of heat through boiler-plates, it was seen that the greater part of the resistance to conduction was credited to surface-resistance. In the transmission of every other kind of energy, viz., sound, light, &c., the passage through the dividing surface between two portions of different kinds of material was obstructed so as to give rise to a difference of energy-potential on the two sides of the dividing surface; and it was only reasonable to suppose that a similar phenomenon occurred in the conduction of heat-energy, giving rise to a temperature-difference. He demurred to the Authors' statement that it had been hitherto universally supposed that the rate of condensation of steam upon a metal surface of lower temperature was infinite. Engineers who, in considering the matter, had remembered Peclet's well-known results in heat-conduction from gases through metal plates to water, must have recognized that in steam-metal heat-conduction there was probably, and almost certainly, a surface-resistance which would make the condensation rate a strictly finite one, however great the steam-metal temperature-difference might be. In one set of experiments only did the Authors appear to have measured the temperature so close as $\frac{1}{100}$ inch to the inside surface, and they did not use the results of these experiments in their subsequent work, relying solely on the readings at $\frac{1}{25}$ inch and greater depths. This was disappointing, and it would be of great value if they would state what instrumental difficulties prevented the continuance of the measurements at $\frac{1}{100}$ inch. It would appear practicable to arrange the contact at this and even a less depth without a blow-through of steam, and he could not think of any other hindrance. The result was that the Authors had to rely upon a somewhat uncertain calculation of surface temperature from measurements taken so far away as $\frac{1}{25}$ inch. On page 20, for Trial XIX, the Authors calculated a maximum surface temperature by adding half the range of 20° F. to the mean wall-temperature, 291° F., obtaining 301° F. But from the diagrams given it would appear that the addition of seven-tenths to eight-tenths of the range to the mean would be a fairer calculation of the maximum. If three-fourths

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[1] Minutes of Proceedings Inst. C.E., vol. xciii. pp. 285–292.

Prof. Smith.

instead of one-half of 20° F. were added to 291° F., since the result was to be subtracted from only 328° F., the resulting difference, namely, 27° F., or only 22° F., would be largely affected. Regarding *Fig. 18*, p. 25, the Authors stated that the small discrepancies between indicator and platinum-thermometer readings were so regular and consistent, that they "cannot be attributed either to errors of the indicator or of the thermometer." This conclusion appeared based upon the vertical differences on the diagram. It would be noticed that horizontally the differences all corresponded closely with what might be due to the well-known horizontal displacement of the indicator-diagram due to the variation of the stretch of the indicator-cord. He had little doubt that the Authors were correct in attributing to valve-leakage a greater relative importance than has previously been given to it. If the valve-leakage depended on deformation of the faces and seats due more to unequal-temperature strains than to mechanical pressure, it would indicate great superiority in this respect in piston over flat slide-valves, and the great importance would also be deduced of equal and opposite ports, so that the flow of steam should be symmetrically divided, and equal temperatures, as far as may be, thus maintained. As the to-and-fro motion of the valve was so important in increasing leakage, it appeared strange that the speed of reciprocation should have no visible influence on it: this point seemed worthy of considerable further investigation. The conditions of heating of a single-acting cylinder were much less complex than those of a double-acting cylinder, and were, therefore, much more favourable to the reliable deduction of conclusions from experimental results; but it might perhaps have been useful to emphasize more than the Authors had in the Paper, that the numerical results for double-acting cylinders must certainly differ very widely from those obtained by the Authors for single-acting cylinders. He thought, in the consideration of this subject, the Paper [1] contributed to the Institution by Messrs. Baraclough and Marks upon their extended experimental work under Professor Carpenter, at Sibley College, a highly important one. This work had been carried out most systematically, and the Paper appeared not to have received the attention due to its scientific character and the amount of new information it afforded.

Prof. Thurston

Professor R. H. THURSTON thought the Paper should be welcomed as the most complete statement as to the condensation of steam in the cylinder that had yet been presented to engineers. The

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[1] Minutes of Proceedings Inst. C.E., vol. cxx. p. 323.

Prof. Thurston. method had been admittedly important, but hitherto unfruitful in most instances. The apparatus used by the Authors would seem to have proved itself exceptionally well fitted to the work; it certainly had been used by skilled and painstaking hands. All information relative to this form of engine waste had been derived hitherto through the study of the steam cycle, the pressure-volume diagram, and the heat-supply. It long ago became obvious to the observer in this field that the temperature and heat-cycle of the metal of the engine cylinder must be observed accurately, as well as that of the steam, to give the needed data for a complete understanding and computation of the heat-wastes of the machine. Some 20 years ago Mr. A. A. Wilson, testing pumping-engines designed by him for the City of Chicago, attempted to secure measurements of fluctuation of temperature of cylinder-wall by the use of thermometers, and, naturally, failed to obtain anything but a record of the mean for the engine-cycle.[1] Before the work of Professor Hall was begun, Mr. Churchill, in the Sibley College steam-engine laboratory, experimented for a long time upon the same subject, using the electric current as his measure of temperature of the metal, and gauging the resistance by a Wheatstone Bridge. He found the temperature constant, apparently, at a depth beneath the interior surface of the cylinder-wall of 0·238 inch, measured a fluctuation of 17° F. at 0·115 inch, and of 25° F. at 0·0426 inch. The speed of the engine was 308 revolutions per minute. The load was light and the steam temperature at entrance was 300° F.[2] A number of other more or less similar investigations had been made; but none had given satisfactory determinations. Those reported in the Paper seemed far more perfect in their exhibition of the internal action of the engine than any hitherto made. It might prove interesting to compare with the diagrams here given, by these later investigators, those obtained in Sibley College laboratories some years ago; Mr. E. T. Adams, the investigator, using a curiously ingenious and strikingly effective system of photographically recording the temperature-cycle.[3] The method consisted, in brief, in the insertion of a thermo-couple at a measured depth into the cylinder-wall, thus following the fluctuations of temperature of the metal. The current of the pile operated a galvanometer of enormously high rate of vibration, and a mirror,

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[1] "Manual of the Steam-Engine," part i. p. 493.

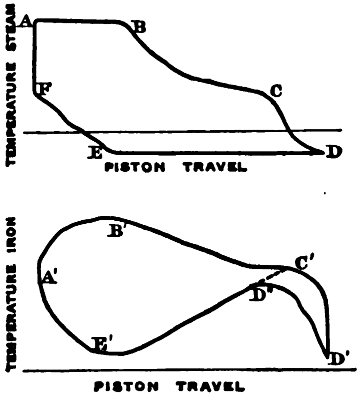
[2] "Cassier's Magazine," 1895, and Transactions of the American Society of Mechanical Engineers, 1895, pp. 445–7.

[3] *Ibid.*

Prof. Thu

fixed upon the needle, reflected a ray of light upon a sensitive plate compelled to move synchronously with the engine-piston. The diagram thus produced was the pressure-volume diagram, as taken with the "indicator," transformed into a temperature-volume diagram; the pair exhibiting the simultaneous variations of temperature and pressure, *Figs. 28.* Many months had been spent by Mr. Adams upon his experiments, and a large proportion of the time was given to the attempt to produce thermo-piles, at once delicate, accurate and durable. These investigations had now been in progress for several years, and their success had encouraged continuance of the work. The diagrams reproduced the changes due to heat-flow $\frac{1}{100}$ inch below the interior surface of the cylinder-wall. The particularly interesting point to be here noted was the singular and unexpected rise in temperature of the metal of the cylinder-head the instant after exhaust had occurred. Whether the diagram thus automatically produced was in any way misleading, or whether it actually represented correctly a curious and unanticipated phenomenon, it was evident that something occurred at that point the revelation of which was likely to throw light upon the working of the engine in this respect. His own explanation was,[1] that the outrush of steam at the opening of the exhaust-valve cooled the face of the metal almost instantly to a minimum temperature; while, immediately after, the current having ceased to flow past with such exaggerated effect in refrigeration, the store of heat just under the surface, in the cylinder-wall, at once raised the temperature again to a new maximum, the cooling taking place again more gradually. Corresponding portions of the steam and metal diagrams were similarly lettered. The dotted line from D′ to C′, *Fig. 28*, showed what, presumably, would have happened had the outflow of steam at exhaust been as slow and smooth as during the return stroke. The rapid cooling period was that evidently during which the water which might have condensed

Figs. 28.



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[1] "Cassier's Magazine," 1895, and Transactions of the American Society of Mechanical Engineers, 1895, pp. 445-7.

rof. Thurston. upon the wall was again vaporizing. A singularly interesting characteristic of the results thus obtained at Sibley College, and also of those presented in the Paper, and one strongly corroborating them, was the close resemblance of the curves produced as their graphical representation to those derived by mathematical investigation as given by Dr. Kirsch as his scientific prophecy of the fact.[1] In his remarkable discussion he gave not only the same general method of variation of the temperature and the same relation of its waves as was shown in *Fig. 13*, but showed substantially the same form of temperature-cycle for the engine as was exhibited in *Figs. 5*, *6*, *et seq.*[2] The temperature-cycle for the metal was found, both by theoretical investigation and by experiment, to be, as would have been anticipated, a reproduction of that of the steam, with, however, the corners rounded off and the amplitude of its temperature-range greatly restricted, and more and more so as the depth of metal became greater. A study of these later results of research, with Kirsch as commentary, was exceedingly interesting, and must aid greatly in clarifying a subject hitherto exceedingly obscure. Kirsch had given the qualitative, and the Authors' investigations now gave the quantitative, determination of results. The one vital datum lacking had hitherto been the relation between the temperatures of the steam and of the cylinder-wall in contact with it. It was in this direction that experiment now largely tended. It seemed somewhat remarkable that 133 years should have elapsed since Watt first measured the internal wastes of the Newcomen engine model of the University of Glasgow,[3] and that it should only now be possible to intelligently describe and approximately measure the fluctuations of temperature in the engine-cylinder to which those wastes were by him attributed. It was 45 years since the late Mr. D. K. Clark,[4] M. Inst. C.E., repeated Watt's measurements on the railway, over 40 years since the great Alsatian, Hirn, measured those losses on stationary

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[1] "Die Bewegung der Warme in den Cylinderwandungen der Dampfmaschine," Dr. Kirsch, Leipzig, Verlag von Arthur Felix, 1889; also Thurston's "Manual of the Steam-Engine," New York, J. Wiley & Sons; and London, Chapman & Hall, Part I, Figs. 141, 142.

[2] *Ibid*, pp. 13*a et seq.*

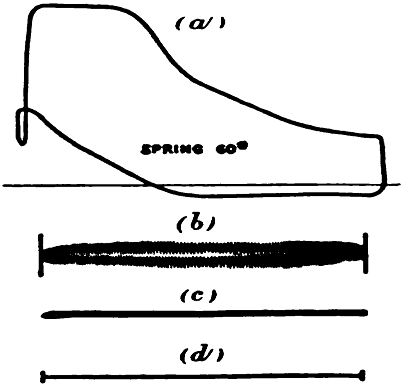
[3] "History of the Steam-Engine," Thurston; New York, D. Appleton & Co., and London, C. Kegan Paul & Co., 1878, p. 85.

[4] "Railway Machinery," London, 1851; "Expansive Working of Steam in Locomotives," Proceedings of the Institution of Mechanical Engineers, 1852, p. 60; Minutes of Proceedings Inst. C.E., vol. lxxii. p. 275.

Prof. Th

engines,[1] and over 30 years since Isherwood,[2] repeated that work on marine engines. So difficult of observation and measurement had been these fluctuating temperatures of the cylinder-wall that all the resources of modern physical science had failed to reveal their precise method and the extent of their variation, until now, except through such academic studies as were made, first, by Professor Cotterill,[3] M. Inst. C.E., and, later, by Kirsch, adopting what might, perhaps, be called the Fourierian system of investigation.[3] Writing of Mr. Adams' remarkable research, Professor Dwelshauvers-Dery concluded[4]: "Ces diagrams confirment les idées que Hirn a émises et que je n'ai cessé de défendre. Ils montrent l'importance qu'il faut attacher à bien protéger le cylindre contre les refroidissements; à en entretenir la haute température par le moyen d'enveloppes complètes à vapeur; a diminuer autant que possible les conduits; enfin à enduire celles de ces surfaces qui ne sont pas exposées au frottement du piston d'une substance, quelconque, huile ou vernis, qui arrête la transmission de la chaleur et augmente la résistance de la couche superficielle du métal, methode preconisée par le Professeur Thurston." As illustrating the latter action, thus advocated, the succeeding figures (*Figs. 29*) showed the results of similar experiments made by Professor A. Barnes, in the course of his work in Sibley College, with the heat-measurements taken at a greater depth within the cylinder-wall, and with a non-conducting coating employed to check the heat-waste. The diagram (*a*) was the steam-engine indicator or pressure-volume diagram; (*b*) was a time-temperature diagram with the cylinder-wall clean; (*c*) showed the effect of simply oiling it; while (*d*) gave the results of coating it with a mixture of oil and

Figs. 29.



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[1] "Théorie Mécanique de Chaleur," Paris, 1876.

[2] "Engineering Researches," 2 vols., Philadelphia, 1863, 1865.

[3] "The Steam-Engine," 1878.

[4] Revue Générale des Sciences, 1895, p. 773.

rof. Thurston. graphite. The most effective system yet employed in carrying this proposal into effect was that of securing first a porous superficial coating and then filling this with a drying oil. This had been found to reduce these heat-wastes very greatly, and to increase the economy of the machine some 10 per cent. or more.[1] This fact might, perhaps, throw some light upon the phenomenon referred to by the Authors, when induced to speculate upon the probable effect of a surface coating of grease. He had found that, by simply wiping such a surface with a greasy cloth, and rubbing as clean as practicable, the surface-conductivity, or resistance, was modified 10 per cent.[2] The oils and greases were almost perfect non-conductors with respect to electrical currents; they were also effective interceptors of heat-currents. The Authors' conclusions and deductions from this admirable investigation seemed to him generally well sustained, both by his own and by other researches, of which they were in turn confirmatory. They computed a heat-transfer of 0·74 thermal units per second (44 thermal units per minute) per square foot per 1° F. temperature range, measured between the steam and the cylinder-wall. His own experiments, of many years ago, gave one-third this quantity per degree of difference between steam and exhaust temperatures, which was not very far different, and Professor Cotterill's results, reduced to the same system of scale, would accord fairly well.

The effect of wetness of steam was probably still an unsolved problem. Experiments at Sibley College, and those of others upon engines with low expansion-ratios, proved it to be unimportant; but it remained to be determined what was the effect of the presence of water in the working steam, with large expansion-ratios and in large engines. It seemed to be the conviction of engineers of experience that its presence was decidedly objectionable in such cases, on the score of deleterious effect upon economy, as well as for other and more obvious reasons. It was certain that the reverse, the superheating of the steam, even but a few degrees, had a decidedly good influence, and effects marked economy. The conclusions of the Authors relative to the effect of varying cut-off (page 45), must be read, having in view the fact that they resulted from experiments upon a small, simple, uneconomical engine, and one in which a three-ported valve was employed which complicated every case by its necessary variation Prof. Thu of the ratio of compression with that of the ratio of expansion. His own formula, there referred to, related to the more usual case of a fairly economical type of engine, with separated steam- and exhaust-valves and no compression, or none variable by the regulating mechanism. He should expect their proposed formula, however, to prove a better expression for use in connection with the now common "automatic" type of small engine. Mr. A. L. Rice, of the Pratt Institute, Brooklyn, had, while at Sibley College, made a very complete examination of the various expressions proposed by investigators to date, and constructed the modified formula—

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[1] "The Final Improvement of the Steam-Engine," Transactions of the United States Naval Institute, 1892.

[2] *Ibid.*

$$y = \frac{C\,(r - I)^a}{d\,N^b},$$

in which a was usually about 0·5, but varied somewhat with pressures, and b was about 0·4, varying somewhat similarly, and C was about 100, and of the values of which constants he gave the figures for a wide range of working conditions, and thus secured practical coincidence with all accepted experimental determinations. The average varied but 0·2 per cent., where Professor Thurston's formula, as employed for many years past, had an average variation of 2 per cent. or 3 per cent.[1] He thought that engineers compelled to deal with such computations would come to the adoption of the simplest form of his own measure, $z = a\sqrt{r}$, finding values of a for the type and class of engine with which they had to deal, and avoiding all other complications. He did not see how the action of leakage could be taken into account. It was purely accidental, without law, varying with different engines enormously, and with the same engine constantly. It might be a serious tax on the engine to-day, and, the engine-driver discovering it, it might be checked and become nil to-morrow. So far as it occurred its effect was practically the same as that of equal initial condensation. The Authors were entitled to thanks for having thrown some light upon its method of occurrence on the process of flow. The reference of the magnitude of the heat-waste to the fluctuation of the now-ascertained temperatures of the cylinder-wall gave important and previously-lacking knowledge of the case. The next essential for the purposes of the designer and of the practitioner was a reliable system of reducing his computations to expressions of relation

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[1] Transactions of the American Society of Mechanical Engineers, vol. xviii. p. 950.

Prof. Thurston. between that variation and the variation of steam temperatures and pressures, in such manner that he might compute probable losses from the indicator diagram, or its ideal as laid down upon his drawing-board. Lacking this connecting link, he had been compelled to adhere to the original form of expression for this waste, as deduced many years ago—

$$y = \frac{a\sqrt{r\,t}}{d};$$

in which the percentage—the fraction, rather—of condensation was measured by a function of r the ratio of expansion, t the time, in seconds, of 1 revolution, and d the diameter of the cylinder in inches; the values of the constant a, for such case, being approximately 4 for the simple engine, unjacketed. External wastes were taken as 0·5 B.T.U. per square foot per hour per degree difference of temperature between steam and surrounding air.[1] The whole subject was one of intense interest, as it was of serious practical importance, and the Authors' investigation must rank among the most famous, as it unquestionably ranked among the most extensive, elaborate, ingenious and fruitful of its class. Its prosecution and its outcome entitled the Authors to the thanks of the profession. They had entered an important and promising field of experimental work.

The Authors. The AUTHORS, in reply to the Correspondence, regretted that, owing to the lateness of the mail, they had had only 2 days to consider the points urged by the various writers. They hoped that they had not in consequence overlooked any important questions. The performance of their engine had been severely criticised in respect of leakage, and it had been suggested that the correctness of their conclusions had thereby been seriously affected. They could not insist too strongly that the fundamental advantage of their method of investigation, as compared with Hirn's analysis, was that the results were entirely independent of leakage. It was also immaterial in their method, so far as the main conclusions were concerned, whether the engines were large or small, single or double-acting, condensing or non-condensing. The leakage of the Robb engine was not abnormal, but compared favourably with that of other engines. They had proved this by experiments under the conditions of running, and not merely by the usual stationary tests, which they had shown to be inadequate. It was

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[1] "Promise and Potency of High-Pressure Steam," Transactions of the *American Society* of Mechanical Engineers, vol. xviii., No. dccxviii.

The Authors.

not fair to take a high-speed engine, with a relatively large balanced valve, and work out its steam-consumption single-acting at one-quarter of its normal speed. Taking the Robb engine, working at one-half cut-off and 250 revolutions per minute, with steam at 100 lbs. per square inch pressure, as fairly representing the average practice in mills where such engines were used; it would develop 90 I.HP., double-acting, non-condensing, with a consumption of about 35 lbs. per I.HP. Only 3 lbs. per I.HP. would then be debited to leakage, which, as they had stated in the Paper, was not an unreasonable proportion for so small and simple a machine.

It was incorrect for Prof. Fidler to suppose that, if the Authors' correction for leakage were applied to the results of Willans and other observers, the missing quantity would disappear wholly from the account. This statement rested on a complete misapprehension, which it was important to remove. The leakage was very different for different valves, but could be approximately estimated by the Authors' formula, p. 32, provided that the fit were the same. For instance, the leak calculated for the valve of the Willans engine in trial $\frac{S\cdot 50}{2\cdot 2}$, assuming the same closeness of fit as in the Robb engine, would amount to between 20 lbs. and 30 lbs. out of a total missing quantity of 90 lbs. By actual experiment, following the Willans method, the Authors had found much larger values of the leakage. They attributed this to wear of the piston-rings, and to leakage of the cut-off ring. Much smaller values might no doubt be obtained by the use of new and tight piston-rings under the conditions obtaining in a factory. That the leakage of a valve in motion was not always a purely accidental circumstance, without law, and constantly varying for the same valve, as stated by Professor Thurston, might be inferred from the Authors' tests, which they believed to be the only measurements hitherto published under conditions which approximated to those of actual running. They had recently re-determined the leakage of the Robb engine-valve in the same manner, after an interval of more than 2 years of constant use, during which the valve had not been scraped or re-fitted, or the engine repaired in any manner. The results were in precise agreement with those given in the Paper.

Hirn's analysis rested on the assumption that steam during expansion, and in rapid vertical motion, followed the laws of statical equilibrium, which was known from other experiments not to be the case. It also depended on difficult and delicate

The Authors. measurements of indicator diagrams, and assumed that the leakage at different points of the stroke was negligible. The Authors, therefore, considered that this method could not be applied with any reason except to specially designed, large, low-speed engines, like Hirn's, where the expansion was slow, and the leakage relatively small. Even in this case they considered that the leakage should be measured under the conditions of running, and that corrections should be applied for it as far as possible at different points of the stroke by some such method as that which they had indicated. Hirn's method of deducing the heat exchange from card measurements had been described by Professor Fidler, who proposed to go further and to differentiate these curves of heat flow. Considering the imperfect and fragmentary character of the original curves, and the variety of possible sources of constant error, there was grave danger that the differentiated curves, if they did not depend upon debatable opinions, would afford wide scope for the exercise of taste in the matter of interpretation. It would be interesting to know whether such curves of $\frac{dh}{dt}$ had been published, and what was the nature of the evidence to be derived from them that the rate of heat-transference did not depend on the temperature-difference. The Authors had themselves obtained decided evidence that the condensation and subsequent re-evaporation of wet steam produced a marked effect in the abstraction of heat from the walls.

With regard to the experiments of Mr. Bryan Donkin and Colonel English on the condensation of steam by the surface-condenser method, it would have been a source of great satisfaction to the Authors if they had been able to deduce from them such a confirmation of their law of condensation as that given by Mr. Maurice Fitzgerald. The conclusions of these able and careful experimentalists had, however, been of a different kind, and their method of calculation did not permit any conclusion to be fairly drawn in favour of the Authors' theory. In the opinion of Mr. Bryan Donkin and Colonel English, "the results of their experiments showed that the film of water deposited by condensation and adherent to a metallic surface resists the transmission of heat in exactly the same way as an equivalent greater thickness of metal would do; so that the difference of temperature between the steam and the actual outer surface of the metal would depend upon the mean thickness of the water-film and upon the rate of flow of heat through it." They had calculated the difference of temperature between the steam and the surface

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of the metal by assuming a convenient ratio of the relative conductivities of water and metal. There was no evidence to justify such a procedure in the case of a layer of drops of water in rapid motion, which could not be treated as if it were a uniform quiescent film of known conductivity. The differences of temperature thus deduced could not, therefore, be used in support of a theory which was directly founded on the assumption of a clean metal surface directly exposed to the steam, but differing from it in temperature, on account of the essentially finite rate of condensation of the steam itself. The Authors, recognising this difficulty, had taken pains to make a special series of experiments by the surface-condenser method,[1] and had endeavoured to prove, by several different lines of reasoning, that such a layer of water-drops could not be treated as a quiescent film, that it did not oppose any material resistance to the passage of heat, and that it might even increase the amount of condensation in virtue of the greater surface exposed to the action of the steam.

The remarkable superheating of the steam close to the walls during compression, as shown in *Fig. 17*, had been verified with five different thermometers, each of which had been independently calibrated. The piston thermometer did not show this effect to the same extent for several reasons. It was in motion, in a wide thin tube, exposed to the air-jacket, and at a lower temperature than the rest of the cover. The cover-thermometer was confined in a narrow hole in the thickness of the cover itself, and the wire was at an average distance of less than $\frac{1}{10}$ inch from the walls. It would appear probable, considering the low conductivity of gases, and comparing the indications of the two thermometers, that this highly superheated surface-layer was very thin, especially on the more exposed surfaces. The Authors could not agree with Mr. Fitzgerald that "it was difficult to conceive how condensation could go on so long as it existed." The superheat was a very small proportion of the total heat, and it could easily be shown by experiment that superheated steam would condense on a metal surface very nearly as fast as saturated steam, provided that the surface were below the condensing temperature. The other suggestion, made by Mr. Fitzgerald, that the dotted area in *Fig. 17*, showing the superheat during compression, ought to be taken for the "condensation area," instead of the saturation curve, which the Authors had invariably made use of, appeared to be somewhat inconsistent with his belief that superheated steam could not condense. The

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[1] Report of the British Association for the Advancement of Science, 1897.

The Authors. adoption of this suggestion would also involve the assumption that steam, if sufficiently superheated, would condense on a surface above the saturation temperature, and that the rate of condensation of steam was, in any case, increased in direct proportion to the degree of superheating.

The Authors had not experimented on the leakage of stuffing-boxes, but failed to see that the tightness of hydraulic apparatus at low temperatures and for excessively slow motion could be urged as an objection to their views on steam-engine leakage. The attempt to make steam glands similarly tight, would either stop the engine, or make the piston rod red hot in a few minutes.

The empirical formula so fully worked out and illustrated by Professor Jacobus, appeared to be capable of representing his own observations better than the Authors'. It would be more interesting, however, if it had a clearer theoretical foundation. It was not easy, for instance, to see why the condensation should be proportional to the range of temperature during expansion rather than to the whole range from admission to exhaust. The former supposition involved the apparent anomaly that, if the steam were carried through the whole stroke, there would be no condensation. A more serious point of difference lay in the effect of variation of speed. According to the results of Professor Jacobus, the missing quantity increased nearly in proportion to the square root of the speed. This might be explained on the Authors' hypothesis by supposing that the variation of speed involved a simultaneous variation in the quality of the steam, which they believed to be a factor of great importance, as explained on p. 41. The diminution of the missing quantity with early cut-off, though somewhat greater than that given by the Authors' formula for the equivalent clearance surface, was in the same direction and of similar magnitude. The formula of Professor Jacobus appeared, however, to represent equally well (*i.e.*, with a range of about 30 per cent. either way) the results of Willans with a totally different engine, in which the missing quantity was nearly independent of the speed, and increased enormously, instead of diminishing, with an early cut-off. No doubt, a part of this increase might reasonably be attributed to increase of leakage with higher pressure, but the conviction of Willans himself to the effect that water was always present in the cylinder was practically conclusive evidence, in the Authors' view, that the condensation in his trials was "limiting" and not "partial." In this case, as explained on p. 52, the condensation should be proportional to the whole range of the steam temperature, an assumption which

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evidently agreed more satisfactorily with the observations. This point had been stated with reference to the results given in the original Paper, but had been omitted in its abridgment.

In the formula quoted by Professor Jacobus no allowance had apparently been made for the difference in duration of exposure of different parts of the surface. The portions of the barrel surface exposed for a very short time were reckoned as of equal importance with the clearance surface. This could hardly be justified theoretically on any view as to the nature of condensation. In the Authors' formula for the equivalent clearance surface allowance had been very accurately made for the varying time of exposure of different parts of the barrel surface, without adding to the complexity of the formula, by the simple addition of the term *ldc* (l = stroke, d = diameter, c = cut-off fraction), instead of writing πldc, the whole area of the surface, as in the formula of Professor Jacobus. A corresponding formula had been given by Professor R. H. Smith in the "final calculation" to which he had referred,[1] in which the portions of the barrel surface exposed up to the time of cut-off, allowing for the time of exposure, had been represented by the expression—

$$\left(1 - \sin\frac{A}{A}\right)\frac{l\,d}{2},$$

where A was the angle turned through by the crank. A Table was added giving the values of this function for different points of cut-off. On working out this formula it would be found that the results were in practical agreement with the simple expression proposed by the Authors up to one-half cut-off. Beyond that point the values of *ldc* were smaller than the true equivalent of the surface exposure. This was intended by the Authors to make approximate allowance for the higher temperature of the parts of the barrel exposed during the latter half of the stroke in a double-acting engine. The formula proposed by Professor Smith did not, as he seemed to imply, make any reference whatever to temperature difference between the steam and the metal, and could not, therefore, be considered as equivalent to the Authors' method of "condensation areas."

No authorities were quoted by Professor Smith for the assumed "surface resistance" due to "difference of energy potential," which had led certain engineers to suppose that the rate of condensation was probably finite. On p. 290 of the article to

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[1] Minutes of Proceedings Inst. C.E., vol. xciii. p. 291.

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e Authors. on a revolving drum. The arrangement was such that the effects of inertia and friction were absolutely eliminated. It took about a quarter of an hour to describe a complete curve, during which time the engine had to be kept running steadily. The apparatus could be locked up and left by itself to write cycle-curves all day if desired, and did not require any interference or any special skill on the part of the observer. It had the advantage of producing a perfectly continuous curve on a larger scale than an ordinary indicator. A complete description of the apparatus, which was much simpler than might be supposed, would be published as soon as the Authors could find leisure for the work. They intended to make a comparison of the curves so obtained under a variety of different conditions with those deduced from the indicator diagram. They might hope in this manner to obtain more definite explanation of the discrepancies they had already observed between the two instruments. The errors of the two indicators chiefly employed in the trials were found to be 3 per cent. too high and 3 per cent. too low respectively. If they had not been corrected their errors would have been of the order suggested by Professor Peabody; but as they were tested hot simultaneously on the same steam against the identical platinum thermometers with which their readings were compared, it is not likely that their residual errors can have been of such a magnitude. The variation of stretch of the indicator cord, as ordinarily observed, would have the opposite effect to that suggested by Professor Smith. In any case, such variation, if it existed, was far too small in the present instance to account for the observed effects. It was, perhaps, also necessary to remark that accurate account had been taken of the obliquity of the connecting-rod in reducing the indicator diagrams to temperature cycle-curves. It appeared probable that there were real differences of temperature existing simultaneously in different parts of the cylinder, and that the temperature of the steam might fall considerably below the saturation point during rapid expansion.

ADDENDUM.

Press Abstract.

At the Ordinary Meeting on Tuesday, the 30th November, Mr. J. Clarke Hawkshaw, Member of Council, in the Chair, the Paper read was on "The Law of Condensation of Steam," by Messrs. Hugh L. Callendar, M.A., and John T. Nicolson, B.Sc.

In the discussion of steam-engine trials it had generally been assumed that the rate of condensation of steam on a surface was practically infinite, so that any surface in direct contact with the steam was immediately heated to the saturation temperature corresponding with the pressure of the steam. It had also been supposed that the amount of condensation under any given conditions was limited, either by the resistance of the film of condensed water to the passage of heat, or by the capacity of the metal or of the circulating water to carry off the heat. In many cases condensation was diminished by films of oil or grease, or by accumulations of air, or by other incrustations or deposits, but these were not considered in the Paper.

The Authors found, on the contrary, as the result of their experiments on a steam-engine running under normal conditions, that a practically clean and dry metal surface was not immediately heated to the temperature of the saturated steam in contact with it, that the rate of condensation of steam was not infinite, but finite and measurable, and that the amount of condensation in any given case was limited chiefly by this finite rate of condensation, and could be calculated in terms of it.

The cyclical variations of temperature in the metallic walls of the cylinder, with each stroke of the engine, were measured by means of thermo-couples inserted at various distances from the inner surface. It was possible thus to deduce the amount of heat absorbed and given out by the metal, and to infer the quantity of steam condensed and re-evaporated at different points of the stroke. The temperature-cycles of the steam were simultaneously measured by a very sensitive platinum thermometer. The observations showed that the temperature of the steam in different parts of the cylinder differed in a systematic way from the saturation temperature as deduced from indicator diagrams.

In order to deduce the condensation from the observed temperature-cycles, it was necessary to determine the conductivity and specific heat of cast iron. A series of experiments were made upon a 4-inch bar of cast iron, and the result found for the conductivity was nearly 30 per cent. smaller than that generally assumed.

At the lowest speed of the experiments, namely, 45 revolutions per minute, the temperature of the surface of the metal at the end of the admission period was found to be never raised higher than within 20° F. of the temperature of the steam, and the rate of condensation at any moment was simply proportional to the difference between the temperature of the steam and the surface. The numerical value found for the rate of condensation was 0·74 B.T.U. per second per square foot of surface per degree F. of difference between the temperature of the steam and the surface. This was equivalent to the condensation of 27 lbs. of steam per square foot per hour at 300° F., for a difference of temperature of 10° F. Assuming this law, the total amount of condensation at any point of the stroke could be inferred by measuring the "condensation areas" on the temperature-cycle diagram, *i.e.*, the areas included between the curves representing the temperature of the steam and of the metal surface.

To compare the results thus found with the missing steam deduced from the indicator diagrams and the feed measurements, the leakage of the valve and piston was determined as nearly as possible under the conditions of running. It was found to be proportional to the difference of pressure and nearly independent of the speed through a considerable range. The usual test for leakage with the valve stationary was found to be of little or no value. From a comparison of leakage tests, it was inferred that a valve in motion, however well fitted, was subject to leakage of a definite type. The leakage took place chiefly in the form of water, by condensation and re-evaporation on the moving surfaces, and

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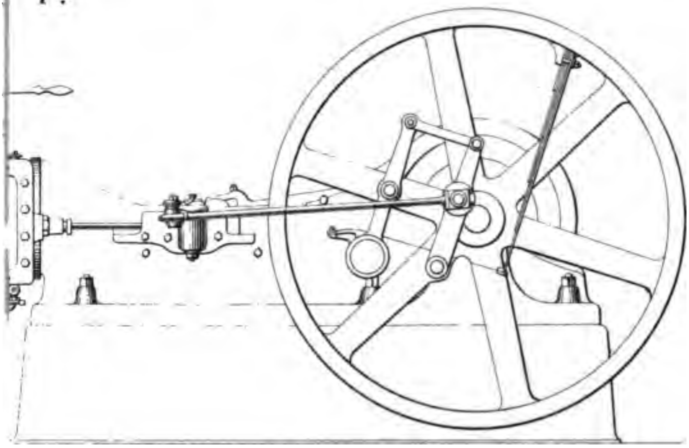

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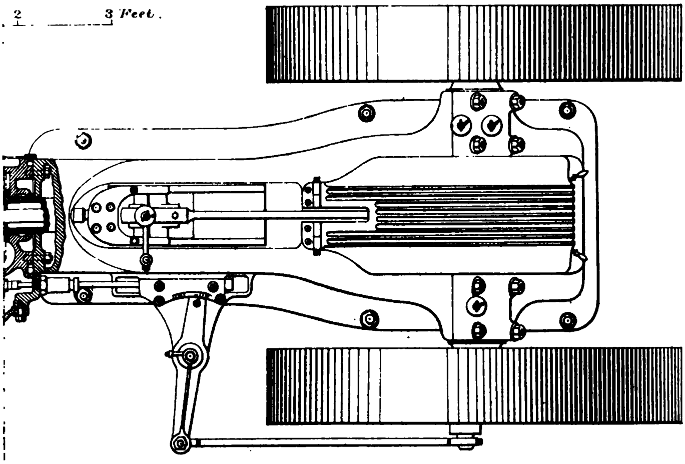

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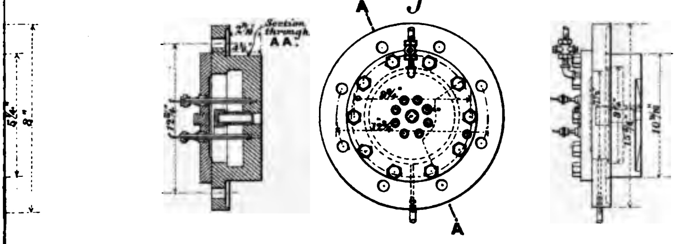

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